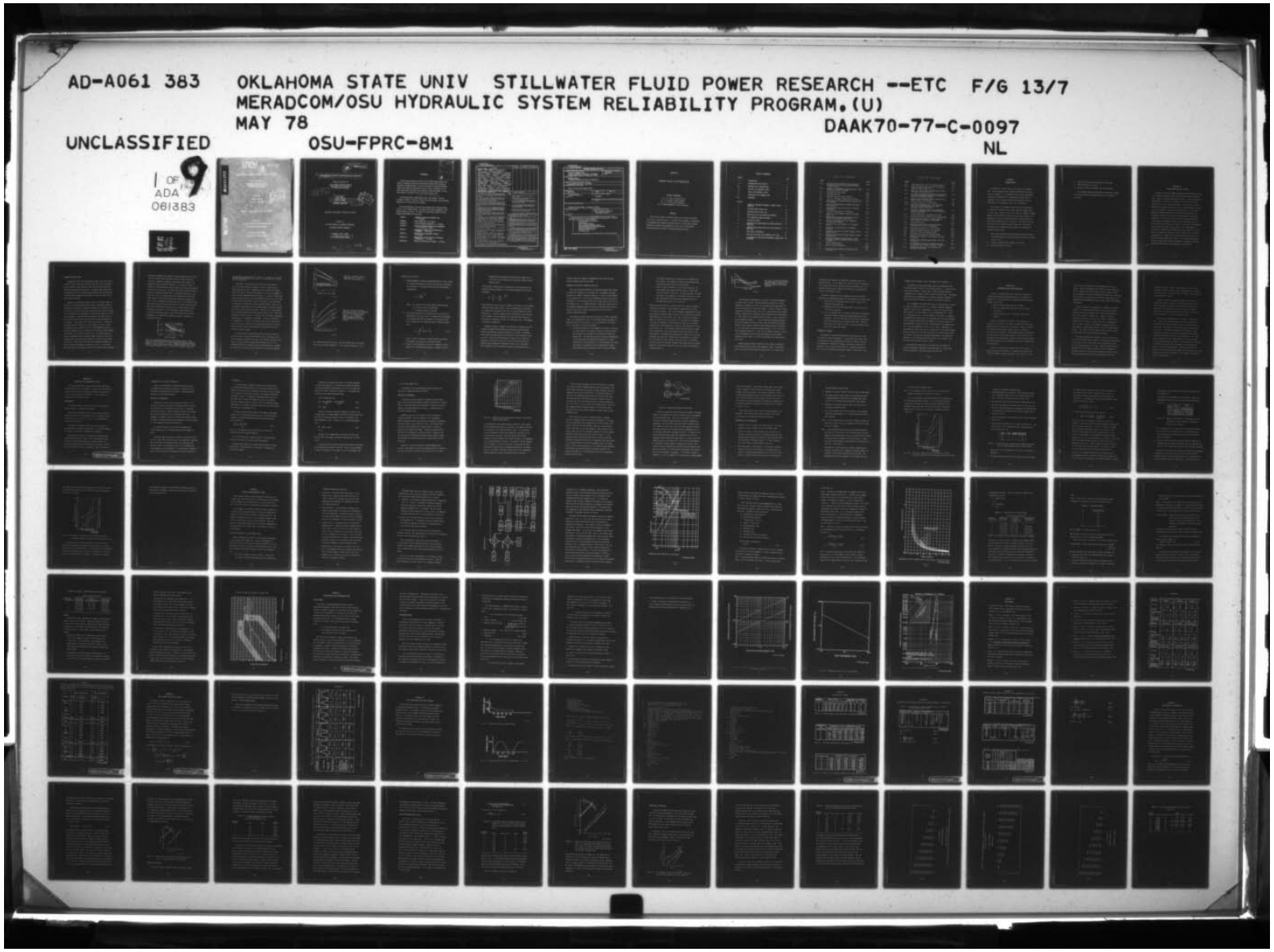
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MAY 78 DAAK70-77-C-0097

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Prepared By Personnel Of

FLUID POWER RESEARCH CENTER
OKLAHOMA STATE UNIVERSITY
STILLWATER, OKLAHOMA

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Fort Belvoir, Virginia 22060

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FOREWORD

This report was prepared by the staff of the Fluid Power Research Center of the Division of Engineering, Technology and Architecture, Oklahoma State University of Agriculture and Applied Sciences. The study was initiated by the Mobility Equipment Research and Development Command, Fort Belvoir, Virginia. Authorization for the study reported herein was granted under Contract No. DAAK70-77-C-0097. The time period covered by this report is from 27 April 1977 to 30 May 1978.

The Contracting Officer's Representative was Mr. Henry Schaefer. The active participation of Messrs. A. Patil and A. Tolbert during critical phases of work contributed significantly to the overall success of the program.

The studies represented by this report were conducted under the general guidance of Dr. E. C. Fitch, Program Director. The details of each study are presented in a self-contained section of this report. The titles of the various sections together with their respective Project Managers are listed below:

SECTION I	**COMPONENT FATIGUE LIFE DETERMINATION – G. E. Maroney**
SECTION II	**NOISE ATTENUATION – G. E. Maroney**
SECTION III	**VALVE CONTAMINANT SENSITIVITY – G. E. Maroney**
SECTION IV	**HYDRAULIC COMPONENT DEGRADATION ASSESSMENT – R. L. Decker and L. E. Bensch**
SECTION V	**MATHEMATICAL MODELING AND MACHINE STATE ASSESSMENT – S.K.R. Iyengar**
SECTION VI	**NONDESTRUCTIVE CONTAMINANT TESTING – R. K. Tessmann**
SECTION VII	**ELECTROSTATIC CHARGE EFFECTS ON FILTRATION PERFORMANCE – L. E. Bensch**
SECTION VIII	**HYDRAULIC FILTER QUALITY CONTROL – L. E. Bensch**

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Reliability Testing Life Prediction Testing Fatigue Testing Endurance Testing Failure Prediction Hydraulic System Diagnostics Component Degradation System Degradation Residual Life Service Life Reliability Contaminant Contaminant Wear Contaminant Sensitivity Ferrography Ferrographic Analysis Hydraulic Pump Gear Pump Hydraulic Valve Filtration Hydraulic Filtration Contamination Control Electrostatic Charge Multi-Pass Filter Test Systems Analysis Performance Appraisal Static Analysis Computer Simulation Mathematical Models Parameter Sensitivity Diagnostics Mobile Hydraulics Component Degradation Semi-Empirical Models State Assessment Filter Filter Quality Control Filtration Ratio Hydraulic Filter Fluid Power Filter Accelerated Testing Endurance Fatigue Hydraulic Noise Sound Measurement Component Sound Levels Airborne Noise Fluidborne Noise Liquidborne Noise Hydraulic Pump Noise Fluid Power Noise Noise Control Hydraulic Noise Control						

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13. ABSTRACT

The overall purpose of the U.S. Army MERADCOM/Oklahoma State University Program is to provide the military with tools for the scientific appraisal of fluid power systems. The activity of the program during this year includes the following:

1. Component Fatigue Life Determination
2. Noise Attenuation
3. Valve Contamination Sensitivity
4. Hydraulic Component Degradation Assessment
5. Mathematical Modeling and Machine State Assessment
6. Non-destructive Contaminant Testing
7. Electrostatic Effects on Filtration
8. Filter Quality Control

This report presents a detailed accounts of the project activities in each of the above areas.

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SECTION I

COMPONENT FATIGUE LIFE DETERMINATION

PROJECT PERSONNEL

Dr. E. C. Fitch, Program Director
Dr. G. E. Maroney, Program Manager
Mr. R. K. King, Project Engineer
Mr. M. Khoddam-Mohammadi, Project Associate

PREFACE

This section presents a detailed account of the project activities in the area of material failure. A specific test procedure is recommended for determining the endurance characteristics of a fluid power component. Guidelines are given for predicting the *"field life"* of fluid power components using the results of accelerated endurance tests.

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CHAPTER I

INTRODUCTION

A component in *"normal"* service will fail due to fatigue or wear. Life discussions without fatigue considerations are incomplete and inadequate. It is essential that accurate engineering techniques be established for determining the fatigue life of components, just as excellent engineering techniques are being established for determining the wear life of components.

The assessment of component fatigue life with a high degree of statistical confidence often requires numerous component tests to *"long"* lives. Because of the crucial nature of component fatigue behavior and the high dollar and time costs of such testing, it is imperative that test and analysis techniques be used which yield the maximum possible amount of information. The objective of this project was to investigate available fatigue test and analysis techniques and to recommend the most appropriate techniques for government specifications.

The following approaches were investigated during the project:

I. Accelerated endurance tests using elevated loads and decreased *"time at load."*

II. Testing to failure versus testing to *no* failure.

III. Sequential endurance testing.

IV. Testing at one load versus testing at varied loads.

V. Weibull analysis of test data.

VI. Automotive industry sequential test specifications.

The following chapters discuss each of these areas. Chapter VI contains specific recommendations based on the results of this study.

CHAPTER II

ACCELERATED FATIGUE TESTING

To reduce the costs and manhours associated with endurance testing, it is desirable to minimize the total test time by using accelerated tests. There are two basic approaches currently in use for accelerating fatigue tests. The first is to decrease the "time at load" of the test cycle (i.e., use a faster cycle rate). The second is to test the component at a higher load than that at which the information is desired (i.e., test a pump at 1.3 times its rated pressure). Both of these methods are indeed successful at reducing the total test time, but they must be used with care and the data obtained must be properly interpreted. If the user falls into the trap of using either method without proper consideration of what the data really mean, the test results can lead to erroneous conclusions. For instance, data analyzed at the Fluid Power Research Center indicate with a high degree of statistical confidence that increasing the "time at load" of the test cycle will decrease the number of cycles to failure [1,2]. Testing the component at a higher load may change the mode of failure to one different from a "normal" field failure. This chapter discusses the effects of using either of these approaches to accelerate testing and presents considerations for interpreting the data obtained.

DECREASING "TIME AT LOAD"

Fatigue tests are often accelerated by using a test cycle having a shorter "time at load" than the actual field duty cycle of interest. Often, field duty cycles are quite long, and tests times can be decreased by using shorter test cycles. In general, most people using this method do not use an interpretation procedure with such data and therefore tacitly assume that the number of cycles to failure is the same for any cycle form.

Contrary to such popular opinion, however, the "time at load" of the test cycle has been demonstrated to have a considerable effect on fatigue life in materials of engineering importance to industry [1,2]. The effect is of special importance to the fluid power industry because it is accentuated by the trapezoidal load wave forms undergone by fluid power components. The effect has been studied at the FPRC since the early 1970's, after it was brought to light by several individuals in the industry. After study of the effect was initiated, a controversy developed within the industry as to the existence of such an effect. In response to this controversy, the U.S. Army MERADCOM sponsored a Component Endurance Study at the FPRC in 1974-75 which provided substantial evidence of the effect's existence [1]. The study analyzed the results of endurance tests on hydraulic pumps, cylinders, hoses, and aluminum alloy test coupons and concluded at the 99.5% level of significance that "cycle time on load" does affect component fatigue life with the effect being greater at longer cycle lives. In 1975, more data were taken at the request of the Basic Fluid Power Research Program

sponsors by fatiguing test coupons in radial bending with both "short" (0.15 sec. time at load) and "long" (1.50 sec. time at load) test cycles [2]. Appendix A summarizes the results of both studies, and Appendix B presents the BFPR coupon data. The coupon test data of Appendix B, resulting from the long and short test cycles, are presented in Fig. II-1 as S-N curves. The trend exhibited by these data follows that of the previously analyzed data; a pronounced difference in cycle lives (up to 3.6 to 1) occurs between data obtained with "short" and "long" cycles, with the effect being more pronounced at higher cycle lives. Thus, a fundamental nonconservative 3-to-1 life difference could exist between mean test data collected with the "short" test cycle and mean field endurance data resulting from the "long" field duty cycle. An important side effect is that laboratories using different test cycles obtain different results, a fact which could prove to be very detrimental to an organization attempting to market a product in a competitive situation.

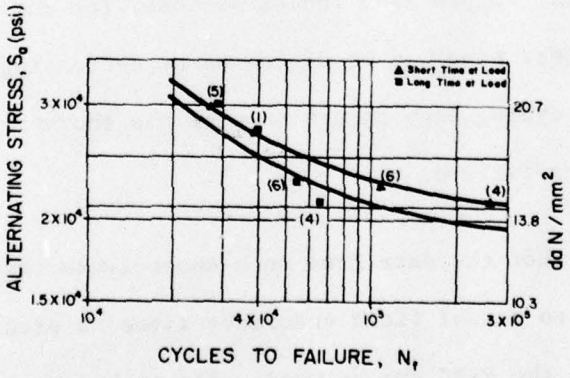
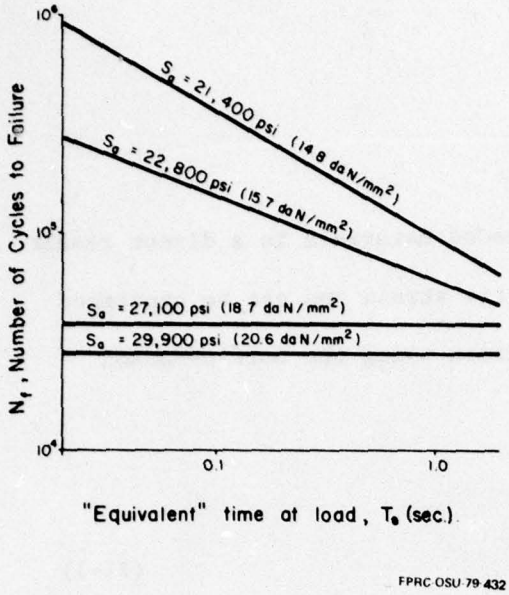


Fig. II-1. Alternating Stress Vs. No. of Cycles to Failure. Data Points and Regression Curves for 2024-T351 Aluminum Coupon Test Data of Appendix A. S_u = 72 ksi, S_y = 52 ksi. Numbers in parenthesis indicate number of coupons failed for each point. Regression curves were fitted to the general equation, $\log S_a = a \log N_f + b + d/\log N_f$. With the

constants being determined to be a = 0.431, b = -3.829, and d = 14.119 for the long cycle data, and a = 0.507, b = -4.436, and d = 15.519 for the short cycle data.

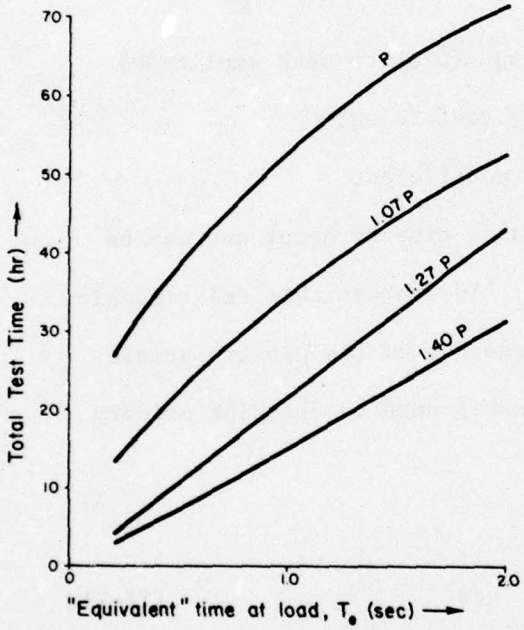
This effect of "time at load" upon cycle life is depicted by Fig. II-2, which presents the number of cycles to failure versus "equivalent time at load." ("Equivalent time at load" is the number used to describe the load-time relationship of a cycle—see Appendix C.) Figure II-2 was developed directly from Fig. II-1, using a straight-line fit (on log-log axes) as a first approximation for N_f vs. T_e, since only two data points are available for each stress level. Note that these data indicate no significant change in life with "time at load" for the two highest stress levels and an increasing change in life with "time at load" as the stress level is decreased. From Fig. II-2, a relationship can be developed between total test time and "time at load." This relationship is presented in Fig. II-3 with the four load levels investigated labeled as P(S_a = 21,400 psi, 14.8 daN/mm^2), 1.07 P(S_a = 22,800 psi, 15.7 daN/mm^2), etc. Figure II-3 indicates that, for any desired load level, the total test time can be decreased by decreasing the "time at load" of the test cycle, even though more of the shorter cycles are required to produce failure.

To interpret what the data from such short-cycle tests actually mean with respect to actual field endurance lives, a prediction technique developed at the FPRC can be used. The technique accounts for the effect of "time at load" by predicting the fatigue life (N_2) of a component subjected to a known wave form (represented by T_{e2}), knowing the results of a fatigue test (N_1) performed on an identical component



FPRC OSU 79-432

Fig. II-2. Effect of "Time at Load" Upon No. of Cycles to Failure for Data of Fig. II-1.



FPRC OSU 79-433

Fig. II-3. Effect of "Time at Load" and Test Load (i.e., pressure) on Total Test Time for Data of Fig. II-1. (P in figure corresponds to S_a = 21,400 psi.) Times determined by assuming "time off load" equal to "time on load."

with another known wave form (T_{el}). The theory underlying the technique rests upon three basic foundations. (For a complete development of the

technique, see Ref. [3].)

1. Fatigue damage of cyclically loaded materials is a direct result of localized (microscopic) plastic strain and can be expressed as a function of the plastic strain using the Coffin-Manson relationship:

$$N_f = \left(\frac{\Delta\varepsilon_p}{2\varepsilon_f'}\right)^{-1/c} \qquad \text{(II-1)}$$

where: $\Delta\varepsilon_p$ = plastic strain range (peak-to-peak amplitude)

ε_f' = fracture ductility coefficient

c = fatigue ductility coefficient

2. This localized plastic strain takes time to occur and can be expressed as a function of time. To express this relationship, it is assumed that the time dependence of the plastic strain can be expressed by the relationship used to describe primary creep:

$$\varepsilon_p = \int_{t_1}^{t_2} C_2 P(t)\, t^{-n}\, dt \qquad \text{(II-2)}$$

where: C_2 and n are material constants and P(t) represents the loading cycle expressed as a function of time.

3. A ratio of lives technique can be used to simplify the above theory into a straightforward prediction technique. This is

accomplished by defining the material failure number, M*, as the ratio of lives resulting from load cycles having different "times on load."

By using these considerations in conjunction with the equivalent time concept outlined in Appendix C, the following expression for M* can be developed:

$$M^* = \frac{N_2}{N_1} = \left(\frac{T_{e1}}{T_{e2}}\right)^{\frac{1-n}{c}} \qquad \text{(II-3)}$$

This technique can then be used to interpret the results of an accelerated fatigue test (N_1) using a "short" test cycle (having T_{e1}) with respect to a predicted field endurance life (N_2) at any known field duty cycle (having T_{e2}). A computer program has been written which uses this equation to predict the field endurance life resulting from a known duty cycle, given a laboratory test life at a known test cycle. This program is presented in Appendix E.

Appendix D presents a summary of the results of verification of the technique performed using component data obtained from BFPR Program sponsors and aluminum coupon data generated at the FPRC. The maximum error between predicted and actual "long cycle" lives indicated by these data is 9%. Note that the data labeled as Coupon Set A in Table D-1 consists of the mean of all of the 15 psi and 16 psi (0.01 and 0.011 daN/mm^2) data of Appendix A (illustrated in Fig. II-1). These data were "averaged" for the verification because the technique does not, at

present, predict the change in the magnitude of the "time at load" effect as longer cycle lives are used.

DECREASING TEST TIME BY INCREASING TEST LOAD

The other widely used method for accelerating fatigue tests is that of testing the component at an elevated load. For example, one might fatigue test a number of hydraulic pumps at 1.5 times its rated pressure (P_r) and assume that, if they survive 100,000 test cycles, the pump will be adequate for the anticipated field duty of one million cycles at the rated pressure, P_r. Or, someone wishing to compare the fatigue lives of two competing components might test them at an elevated load in order to obtain the data sooner (since the numbers are only to be used for relative comparisons anyway).

This approach should be used with caution due to several considerations. One consideration is that, to intelligently make a statement of equivalence such as "10^5 cycles at 1.5 P_r is equivalent to 10^6 cycles at P_r," one needs to have prior knowledge of:

1. S-N curve information for the component in question. The slope of the S-N curve in the region of interest is needed to determine how much the load needs to be increased to produce the desired life. This information allows a rational interpretation of the data so that the conclusions reached are not overly conservative or overly optimistic. S-N curves gained from coupon tests may not be adequate for this interpretation, however, since shapes, stress concentration factors, types of loading,

and material properties of internal parts of components may not be known. This implies that some of the components must previously have been tested to the long lives.

2. Mode of failure information for the lives of interest. The same component may undergo different modes of failure when tested at different load levels. Failures may appear in different parts of the component or in different locations of a part. This consideration also implies that components must have previously been tested to the long lives.

These factors make the use of elevated loads for A versus B component comparison tests risky. Several things can happen which could lead to an invalid conclusion. One or both components undergoing such a test may experience different modes of failure at the elevated load than they would experience at the "normal" load. Even if the modes of failure remain the same for the two conditions, if the slopes of the component S-N curves are different, the curves may cross and give an erroneous indication as to which component is superior. This condition is illustrated in Fig. II-4. As an example, consider the case where two components, represented by the curves labeled "strong" and "ductile," are being compared. The design load corresponds to a stress of S_1, but the components are tested at an elevated load corresponding to a stress of S_2 to accelerate the test. Clearly, this test leads to an erroneous conclusion, since the "ductile" component lasts longer than the "strong" component at the elevated stress, S_2; but, the results are reversed at the design stress, S_1.

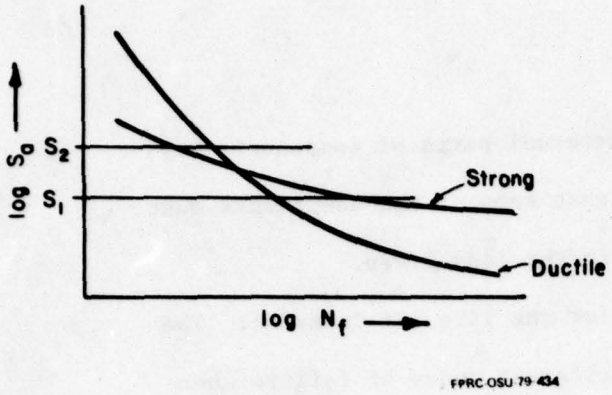


Fig. II-4. S-N Curves Illustrating "Cross-Over" of Low-Cycle and High-Cycle Fatigue Behavior of Components Made of Two Different Materials.

The items under investigation do not have to be complex components for these curves to cross. Desirable material properties for fatigue resistance are different for low cycle fatigue (failure before about 10^4 cycles) than they are for high cycle fatigue. For good low-cycle fatigue resistance, the material needs to be able to resist large plastic strains through a high ductility, requiring a high fatigue ductility coefficient, ε_f', which implies a low fatigue ductility exponent, c, and a high cyclic strain hardening exponent, n'. For good high-cycle fatigue resistance, the material needs to be able to resist large stresses without yielding through a high strength, requiring a high fatigue strength coefficient, σ_f', which implies a low fatigue strength exponent, b, and a low cyclic strain hardening exponent, n'. Thus, the two fatigue resistance criteria are not complimentary, and materials having good low cycle fatigue resistance are not expected to also have good high cycle fatigue resistance [4,5].

Another common situation in which S-N curves might be expected to cross arises from the fact that many components consist of both parts made from a material exhibiting an endurance limit (i.e., steels) and

parts made from a material not exhibiting an endurance limit (i.e., aluminum alloys). If, for example, the critical location of Component A is on a steel part and the critical location of Component B is on an aluminum alloy part with a higher static strength, the component S-N curves can be expected to cross.

The above considerations can be summarized into the following guidelines for interpreting fatigue life data obtained at elevated loads.

1. Should have prior knowledge that the mode of failure is not changed by increasing load.
2. Should have prior data indicating slope of S-N curve for the life region in question.
3. A vs. B comparison tests at elevated loads should be approached with great caution, following the above guidelines for both components. Do not use load levels which will give lives which are too far removed from the expected field lives. Stay well above lives of 10^4 cycles in order to avoid the transition from low cycle to high cycle fatigue.

COMPARISON OF METHODS

The method of accelerating fatigue tests by decreasing the "time at load" offers several advantages. It accelerates tests greatly, and a straightforward data interpretation procedure for similar loads has been presented. No evidence of changes in the mode of failure due to changes in "time at load" has been observed. Information needed for use of the data interpretation procedure consists of values for c, the

fatigue ductility exponent, and n, the dynamic strain exponent.

The method of accelerating fatigue tests by increasing the load level also allows for tests to be accelerated greatly. It should be used with caution, however, to insure that it does not provide erroneous information due to changes in mode of failure with load level. Information needed for proper interpretation of such data consists of knowledge of the S-N curve slope and modes of failure in the region of interest.

For an example of the amount that the total test time can be reduced by these two methods, see Fig. II-3. Consider a hydraulic component expected to have a life corresponding to a total test time of 72 hours when tested at its rated pressure, P, at an "equivalent time at load" of 2.0 seconds per cycle. If this component's S-N curves are adequately represented by the data of Fig. II-1 (from which Fig. II-3 was developed) and if the mode of failure does not change, the total test time can be decreased from 72 hours to 31 hours by increasing the test pressure from P to 1.4P while keeping the 2.0 sec. "time at load." Conversely, by decreasing the "time at load" from 2.0 seconds to 0.2 seconds and keeping the test pressure at P, the total test time can be reduced from 72 hours to 26 hours. By using both approaches simultaneously, increasing test pressure to 1.4P, and decreasing "time at load" to 0.2 seconds, total test time could be reduced to only 4 hours.

The concepts developed above were presented to the Society of Automotive Engineers Earthmoving Industry Conference in a paper, "Two Practical Approaches to Minimize Fatigue Test Time" [6].

CHAPTER III

ENDURANCE TEST CONSIDERATIONS

A number of considerations must be taken into account when formulating an endurance test specification for a component. In addition to the items previously discussed concerning accelerated testing, they include:

1. Tests to failure versus tests to no failure.
2. Tests at a single load level versus tests at multiple load levels.
3. How many samples to test. Fixed sample size versus variable.

The first consideration is whether to test components to failure or to no failure. This choice is equivalent to the choice between testing by Attributes or by Variables. Testing to failure is Variables Testing because a number (cycles to failure) is obtained for each component. This information about the component's life distribution, when analyzed properly, allows decisions to be made using smaller sample sizes than needed with attributes testing. The actual reliability of the components can also be estimated from test information.

Tests to no failure are attributes tests in that each test gives only pass/fail information. If components last longer than a specified number of cycles, they pass; if not, they fail. Such

tests do not allow estimation of the component's life distribution. Since each specimen does not furnish as much information as in a variables test, larger sample sizes are needed to attain the same degree of statistical confidence in predictions. No failure tests can result in shorter total test times in some instances if components are overdesigned and lives are much longer than specified requirements.

Chapter IV presents a comparison of attributes and variables testing. It also determines the sample sizes required to give a specified statistical confidence for an example using both attributes and one type of variables tests. Chapter V includes a discussion of tests to no failure as the *"default position"* in a sequential test. Using this concept, this test is conducted as a test to failure, but once a given number of specimens survive a given number of cycles without failure, the test is halted without failure. This occurs only when components have a mean life which is much greater than the specification requirement.

The second consideration is whether to test at a single load, or to test at multiple loads. Chapter II discussed the ramifications of the choice of load (*"field duty"* load versus elevated loads), but only considered tests at one load. Analysis methods have been investigated for both types of testing. Chapter IV presents an analysis technique for multiple load testing using response curves. This method analyzes the statistical distribution of the load

required to produce failure at a given number of cycles. This information is used to predict with a known statistical confidence, the load at which a given percentage of components will survive the specified number of cycles.

Chapter V presents an analysis technique for single load testing using Weibull analysis. This method analyzes the statistical distribution of fatigue lives resulting from testing at a given load. This information is used to predict with a known statistical confidence, the number of cycles which a given percentage of components will survive at the given load.

The third major consideration is the number of specimens to test. Despite the tremendous importance of the decisions which are based on test results, sample sizes are often chosen without a great deal of thought and little appreciation of the risks that are involved. Both analysis techniques described above include statistical techniques for determining the sample sizes required to obtain the desired confidence in test predictions. Both plans employ sequential testing. Using sequential testing, the number to test is not determined in advance. Instead, statistical analysis is performed after each test, to determine if the desired confidence in results has been attained. Testing is continued until either the desired confidence is attained, the tester decides to cease testing and accept lower confidence, or the component is clearly not going to pass the test.

CHAPTER IV

MULTIPLE LOAD SEQUENTIAL TESTS

This chapter describes a fatigue test plan which uses tests at more than one load. It presents a sequential test plan in which response curves are used to optimize the sample size required to attain a desired statistical confidence. It also presents for comparison an attributes test plan.

APPROACHES

In any test procedure, there are two ways of describing the measured quality: attributes and variables.

An attribute measurement is one in which a quality characteristic falls into a limited number of discrete categories (such as white or black, off or on). The *"go/no go"* gauge provides a measurement by attributes.

Measurements by variables involve the use of a scale that can theoretically be infinitely subdivided. The length and weight of an object are examples of measurement by variables.

Measurement by attributes is generally less expensive than measurement by variables. However, many more measurements must be taken to obtain the same amount of information than can be obtained by a small number of measurements by variables. Measurements by attributes is unable to express the range of agreement or

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disagreement with a certain requirement.

Some of the techniques of describing attribute tests are the binomial and chi-square distributions. Response curves are one method of analyzing tests by variables. These methods are described in the following sections.

TESTING BY ATTRIBUTES

Components are often fatigue tested by attributes. The attribute in question is whether the component survives or fails when subjected to a given number of test cycles. The binomial distribution provides a means of analyzing attribute test results. The binomial distribution gives the probabilities that 0, 1, 2,··· n members of a sample of size n will possess some attribute, when the sample is randomly selected from a population in which a portion "p" of the members possess this attribute [8].

To illustrate the use of this method to determine the required sample size, the following fatigue testing example is given.

A hydraulic pump is required to operate at a pressure of 300 bar for at least 10^6 (1 million) cycles. The consumer has specified that at least 95% of the pumps must meet this requirement (95% assurance level). He is willing to accept a 2.5% consumer risk (corresponding to a 95% confidence). A certain number of pumps are tested. Upon testing, all the specimens were able to satisfy the

requirements.

If a given number of pumps are tested and no failures occur, the binomial distribution provides a 95% confidence interval on the number of failures that can be expected if the same test were repeated. If ten pumps are tested with no failures, the binomial distribution presented in Table F-1 of Appendix F indicates that we can be 95% confident that 0 to 27% of the samples will fail. This implies that more specimens need to be tested to meet the given requirements. From Table F-1, the number of specimens that must be tested to insure 95% confidence is approximately 100.

To obtain a more accurate estimate of the number of specimens required, the chi-square distribution is used. Chi-square (X^2) is a measure of the deviation between observed and expected values and is defined as:

$$X^2 = \sum \frac{(f_n - F_n)^2}{F_n} \qquad \text{(IV-1)}$$

where f_n is the sample count of individuals who do and do not possess the attribute being investigated.

F_n is the hypothetical or expected number of individuals who do and do not possess the attribute being investigated.

Relatively high values of chi-square indicate relatively large discrepancies. The significance of chi-square is illustrated in the following example.

Applying the chi-square principle to the problem explained in the previous section, it is necessary to estimate the sample size that can generate 95% confidence that 95% of the pumps will survive 10^6 cycles (1 million) at 300 bars. It is assumed that no failures were observed during testing.

Let the sample size be n.

$$X^2 = \frac{(h - 0.95n)^2}{0.95} + \frac{(0 - 0.05n)^2}{0.05n} \qquad \text{(IV-2)}$$

$$X^2 = 0.053n \qquad \text{(IV-3)}$$

From the chi-square distribution presented in Table F-2 for a single degree of freedom (number of possible attributes minus one), the value of chi-square required to give a 5% consumer's risk (95% confidence) is 3.84. From Eq. (IV-3), the required sample size:

$$X^2 = 3.84 = 0.053n \qquad \text{(IV-4)}$$

$$n = 72$$

Therefore, the 72 pumps have to be tested to be 95% confident that 95% of the pumps can operate at 300 bars for 1 million cycles.

The binomial and chi-square distribution have given the estimate of the sample size as between 72 and 100. Thus, if the measurement is done by attributes, it is necessary to test 72-100 pumps, which

is a very large sample size.

To determine if this large sample size can be reduced, the next section analyzes testing by variables.

TESTING BY VARIABLES

In testing of hydraulic pumps by attributes, the only information used was whether the pump can operate at 300 bars for 1 million cycles. The quantity which was an indicator of success was discrete (survive a specified number of test cycles). These variables are related by response curves.

Response curves are a very useful tool in the analysis of fatigue data [9,10]. They are estimates of the relationship between operating pressure levels and the percentage of a population that would survive a given number of cycles. The response curves are usually drawn assuming that fatigue pressures to produce a given life are normally distributed. Fatigue test experience has shown that this assumption is usually valid [9]. Response curves are often used for evaluating the median fatigue strength (rating) and estimating the pressure required for a given percent survival of N cycles.

Figure IV-1 relates the percent of tested components that survive a given number of cycles to the applied stress (or pressure). Note that this curve is valid for only one cycle life of interest.

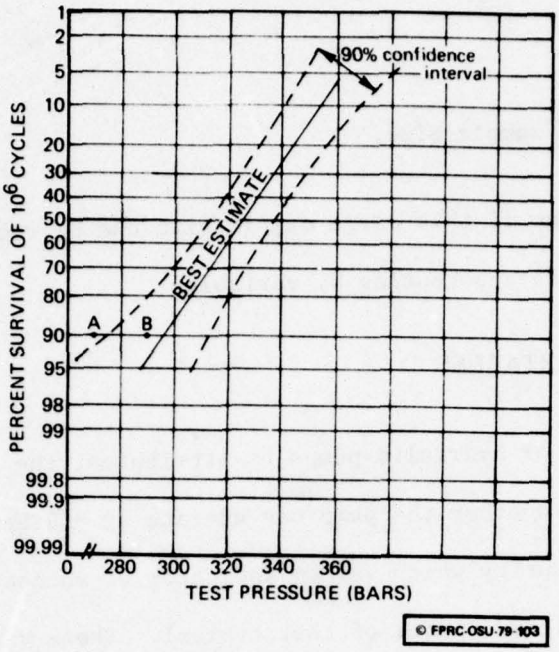


Fig. IV-1. Response Curve Relating Percent Survival of One Million Cycles to Test Pressure.

If the assumption that fatigue pressures required to yield a given life are normally distributed is a good one, the response curve will be a straight line on normal probability paper. It can therefore be easily estimated by using linear regression techniques on test or field service life data. The line obtained from the data represents the best estimate of the percent survival of the component (of the required number of cycles). For instance, from the response curve of Figure IV-1, the *"best estimate"* in predicting the field lives would be that 94% of all components would survive one million cycles at a pressure of 290 bars, while only 14% of all components would be expected to survive one million cycles at 350 bars.

Since the test data points will not lie directly on a straight line, they will have a standard deviation about the regression line. This standard deviation can be used to discuss confidence intervals (bands) about the *"best estimate"* line. For instance, Figure IV-1 contains a 90% confidence interval about the *"best estimate"* line. This indicates that we have 90% statistical confidence, from the test data, that the actual values of percent survival estimated by the *"best estimate"* line lie within this confidence band. Note that, for a symmetrical 90% confidence interval, 95% confidence exists that the actual value of percent survival is greater than the lower confidence limit.

This concept can be used to determine whether a component *"passes"* an endurance test with the desired confidence. If to meet the test specification, the component must have 90% survival of 10^6 cycles at 270 bars with 95% confidence, then the test data must be *"better"* than point A, with 95% confidence, This criterion is met in this case, since the 90% confidence interval for the test pressure required to produce 90% survival lies above this point. If the test specification had required 90% survival at 290 bars with 90% confidence (represented by point B), the components would have failed to meet specifications, since the 90% confidence band is not above point B. These concepts pave the way for a sequential test procedure to be defined, such that the number of specimens required to be tested is determined by the relationship between the test specifications and the test data.

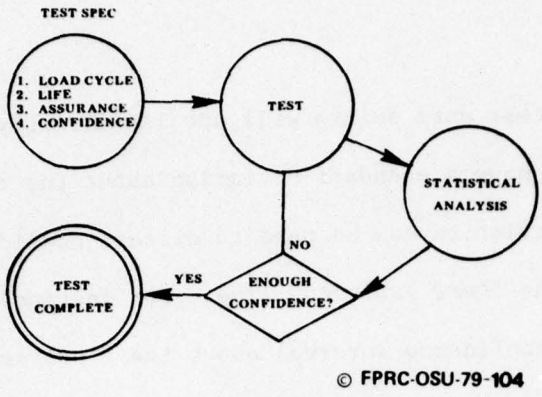


Fig. IV-2. Sequential Endurance Test Procedure.

The use of a sequential test plan allows the number of specimens tested to be minimized for a desired confidence level corresponding to a given consumer risk. Such a plan is outlined in Fig. IV-2. First the test specification defining an acceptable component is determined. The test specification must define acceptable values for: number of cycles, test cycle wave form (pressure and time at pressure), assurance level (percent defective components that can be tolerated), and desired confidence level [7]. For instance, the test specification might require 95% confidence that 90% of all components would withstand 10^6 applications of a 270 bar load cycle (plotted as point A in Figure IV-1). After the test specification is determined, a minimal number of components are tested. Then these test data are statistically analyzed using the response curve method to determine if the specification requirements have been met with the desired degree of confidence. If they have, testing is terminated. If not, the sample size is increased by a minimal amount,

and the test repeated. Testing with a larger sample size provides a better estimate of the true survival distribution and yields a narrower confidence interval. This narrower confidence interval provides more confidence that the actual distribution will be *"better than"* any specified point (such as points A and B in Fig. IV-1). This process of testing, analyzing, and comparing is repeated until the desired confidence is gained that the specimen either *"passes"* or *"fails to pass."*

These ideas form the basis for the following sequential test procedure. The procedure uses equations from Ref. [10], which are presented in the example following the procedure.

SEQUENTIAL TEST PROCEDURE

1. Determine critical values for test specification: load cycle, number of cycles, assurance level, and confidence level.
2. Select five groups of three specimens from the population of interest. Five groups of three is the minimum sample size that produces usable estimates for relatively high confidence levels. (See discussion in following example.) If lower confidence levels will suffice, fewer specimens can be used.
3. Test these groups at five pressures. The pressures should be selected so that the test pressure for the middle group will produce an anticipated median fatigue life to the cycle life of interest. Reference [9] provides guidelines for choosing the

interval between pressure levels.

4. Determine the percent survival for the test data of each group.

5. Plot these percent survival versus test pressure data points on normal probability paper and fit a straight line (*"best estimate line"*) to them using the method of least squares.

6. Determine the confidence interval about the fitted line for the desired confidence level.

7. Plot the acceptable point from the test spec on the response curve chart and determine if it is within or below the confidence interval.

8(a). If the acceptable point is below the confidence interval (as was point A in Fig. IV-1), the component has *"passed"* the test. The test is complete.

8(b). If the acceptable point is within the confidence interval, but below the *"best estimate"* line, the data predict that the component will probably pass the test, but have failed to provide the desired level of confidence in this prediction. In this case, test one more sample at each pressure, and reanalyze the test data. (Go to step 3.)

Alternatively, if the resources to continue testing are not available, one might wish to terminate testing and compute the confidence level actually indicated by the test data. The component would then be considered to have *"passed"* the test,

but at some lower confidence level.

8(c). If the acceptance point is above the *"best estimate"* line, consider the component as having failed the test.

Use of the procedure and the needed equations are presented in the following hypothetical example. A lot of prototype hydraulic valves has been produced and requires testing to determine if 95% confidence exists that 85% of the valves will survive one million applications of a given test cycle having a maximum test pressure of 380 bars.

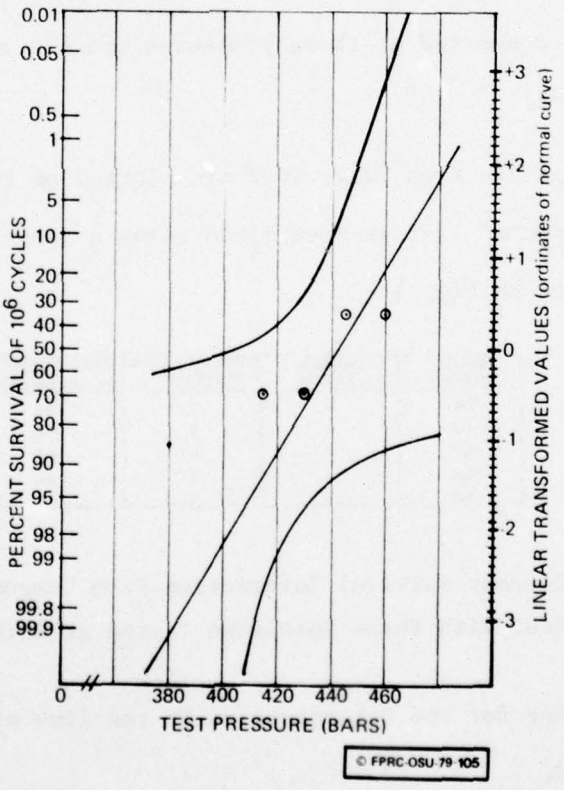


Fig. IV-3. Data Points, Regression Line, and Confidence Interval for Example Problem with Three Specimens Tested at Each Load.

Following the procedure presented above:

1. The acceptance point is defined as: 95% confidence that 85% of the specimens (valves) survive one million cycles at 380 bars. This point is plotted as point S on Fig. IV-3.

2. Five groups of three valves are chosen at random from the lot.

3. The five test pressures are chosen as 400, 415, 430, 445, and 460 bars, based on previous test information that indicates that a test pressure of 430 bars produces a median fatigue life of one million cycles.

4. Tests are conducted at these pressures and the results tabulated (Table IV-2).

5. The data points from Table IV-2 are plotted on Figure IV-3. The *"best estimate"* line is determined using a least squares fit and plotted on Fig. IV-3.

APPLIED PRESSURE	NUMBER TESTED	NUMBER SURVIVING ONE MILLION CYCLES	PERCENT SURVIVAL OF ONE MILLION CYCLES
400	3	3	1.00
415	3	2	.67
430	3	2	.67
445	3	1	.33
460	3	1	.33

Table IV-2. Percent Survival Information from Component Endurance Test with Three Specimens Tested at Each Pressure.

Calculations for the determination of the line are presented in Appendix G.

6. The 90% confidence interval for the response curve is determined.

The computation here assumes that the normal transformation of the percent survival value is suitable, that is, that the transformed percentages are normally distributed about the fitted line. The standard deviation of the transformed y values about the fitted line is [10].

$$s = \left(\frac{\Sigma (y-y_f)^2}{k-2} \right)^{\frac{1}{2}} = 0.81 \qquad \text{(IV-5)}$$

The symmetrical confidence limits are given by:

$$y_f \pm s \left[2 F (2,k-2) \right]^{\frac{1}{2}} \left[1/k + \frac{(x-\bar{x})^2}{\Sigma (x-\bar{x})^2} \right]^{\frac{1}{2}} \qquad \text{(IV-6)}$$

where F (2, k-2) is evaluated from Table F-3 for 2 and k-2 degrees of freedom for numerator and for denominator, respectively. Since the value obtained for F is directly related to the width of the confidence band, small values of F are desired. From Table F-3, with numerator degrees of freedom fixed at 2, the value of F rises sharply as the denominator degrees of freedom decrease to one. For low probabilities of a larger F, (0.10, 0.05, 0.01), this sharp increase is seen to "break" at denominator degrees of freedom of 3. Thus the number of groups to be tested (k) was chosen as five (so that k-2=3). For this test, F(2,3) for the determination of the 90% confidence band is 5.46.

The effect of the standard deviation on the formula is also very important because it is directly related to the sample size. As the sample size is increased, the standard deviation is expected

to decrease, since the data points are able to more closely fit the straight line.

Calculations for determining the 90% confidence interval are summarized in Table IV-3 and the resulting interval plotted on Fig. IV-3.

APPLIED PRESSURE (BARS)	CODED VALUE, x	$(x - \bar{x})^2$	$A = \frac{(x-\bar{x})^2}{\sum (x-\bar{x})^2}$	$B = (1/k + A)^{1/2}$	$B_S = (2F_{2,3})^{1/2}$	Y_f	90% CONFIDENCE LIMITS $Y_f \pm B_S \cdot (2F_{2,3})^{1/2}$			
							y VALUES		p VALUES	
							UPPER	LOWER	UPPER	LOWER
400	1	4	0.40	0.77	2.06	-2.12	-0.06	-4.18	52.4%	0.0%
415	2	1	0.10	0.55	1.47	-1.36	0.11	-2.83	45.6	99.77%
430	3	0	0.0	0.45	1.21	-0.60	0.61	-1.81	27.1	96.49%
445	4	1	0.10	0.55	1.47	0.16	1.63	-1.31	5.16%	90.49%
460	5	4	0.40	0.77	2.06	0.92	2.98	-1.14	0.14%	87.29%

Table IV-3. Summary of Calculations for Determination of 90% Confidence Interval of Example. (Three Valves Tested at Each Pressure.)

7. The acceptable point S is compared with the confidence interval. For this case, S is within the interval, indicating that the test does not meet the acceptance criteria with the required degree of confidence.

8. Since the original test did not indicate that the component was acceptable with the required confidence, one more component (total of four) at each stress level is tested and the results reanalyzed.

Hypothetical test data and the necessary calculations are presented in Appendix H. Data points, the *"best estimate"* line, and 90% confidence interval line are plotted in Fig. IV-4. Notice that here the 90% confidence interval is above point S. This indicates that

the required confidence has been obtained that the specified criteria, 85% survival of one million cycles at 380 bars, have been satisfied. The test is terminated.

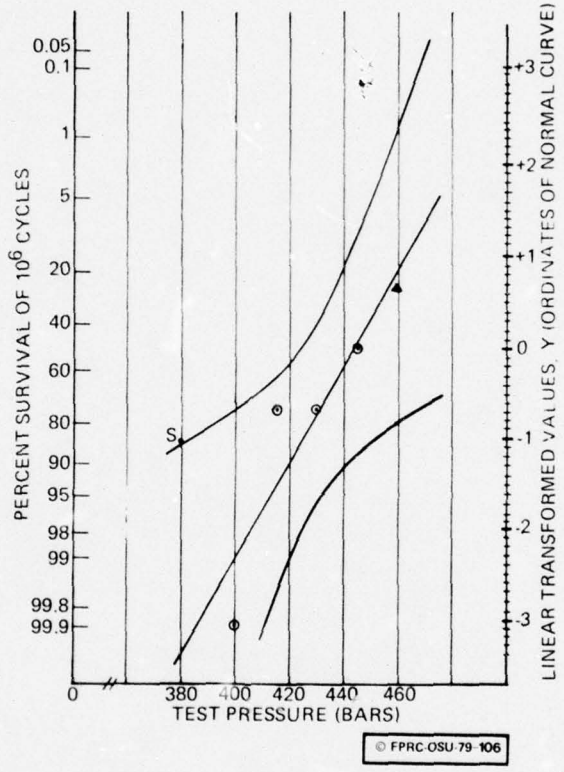


Fig. IV-4. Data Points, Regression Line, and Confidence Interval for Example Problem with Four Specimens Tested at Each Load.

An important aspect of this test method is that the required width of the confidence band depends on the proximity of the actual distribution to the specified point S. This means that if the component is substantially stronger than required (overdesigned), it can pass the test with a small number of specimens. Alternately, a component

only marginally stronger than specification requirements requires a large number of specimens tested to give the desired confidence that the requirements are met.

CHAPTER V

SINGLE LOAD SEQUENTIAL TESTS

This chapter presents a plan for sequential fatigue tests at a single load. These tests make use of the fatigue life distribution of the components at a single load level. This load level can be the *"field duty"* load or an elevated load, as discussed in Chapter 2.

This approach to testing was studied by project personnel and a sequential test plan was developed. This plan was presented in a paper at the 1977 National Conference on Fluid Power [11]. Subsequent to formulation of this plan, an automotive industry sequential test specification (Ford) was discovered [13]. This plan provides a more complete approach and is therefore recommended over the plan presented in reference [11]. The Ford plan is outlined in this chapter.

Critical Parameters for Fatigue Tests

Several parameters are critical to the thorough understanding of the results of any fatigue test. Tests are performed for a variety of reasons, but all seek to determine information about the distribution of lives as a function of the applied load cycle. One set of parameters which must be known to provide this information are:

1. Cycle Life: Number of cycles to failure. These values will have a statistical distribution for any applied load

(component endurance distribution).

2. Load Cycle: The pressure-time characteristics of the applied load. Previous studies have shown that the *"time at pressure"* of the load cycles as well as the maximum pressure amplitude is a significant variable for fatigue failure [1].

3. Percent Survival: The percentage of components that survive a given number of load cycles, as determined by the life distribution. The familiar concept of the B_{10} life specifies the number of cycles for which the percent failures, the complement of the percent survival, is 10%.

4. Assurance Level: The maximum percent of unacceptable components that is considered tolerable (roughly equivalent to AQL). Since we wish to test a small sample to accept or reject a larger lot, we can never be sure that all of the components are acceptable. If we can specify a tolerable percent defective, however, statistical theory provides an estimate of the probability that the percent of defective components is within allowable limits.

5. Consumer's Risk: The chance of accepting a *"bad"* lot. This is the maximum risk that the consumer takes in using tests of small samples to qualify larger lots. The consumer's risk is directly related to the statistical concept of a confidence interval.

Although fatigue tests are conducted to meet a wide range of objectives, most, if not all testing of fluid power components can be classified under two basic motivations for testing:

1. Acceptance: (of a design, a manufacturing process, a production lot, etc.). The test is performed to determine compliance with a test waveform, required life, percent survival and designed confidence level. For example, "The test should indicate with 95% confidence that the B_{10} life of the component be greater than 10^6 cycles of the given 3000 psi load cycle."
2. Rating: Here the motivation is to use the test to help write the component spec: to determine at a given confidence level what loads the component can sustain for a given number of cycles with a given percent survival.

The techniques presented in this paper were developed for acceptance testing. The relationships of the critical test parameters for this case are illustrated in Fig. V-1.

The test procedure is based upon a test specification. To be truly definitive, the spec should specify the test waveform, required life, consumer and producer's risks, desired confidence level, desired Weibull slope, and the maximum number that will be tested.

The test load cycle, when applied to the specimens, produces a distribution of percent survival versus cycle life. This test data distribution is a direct result of the component endurance

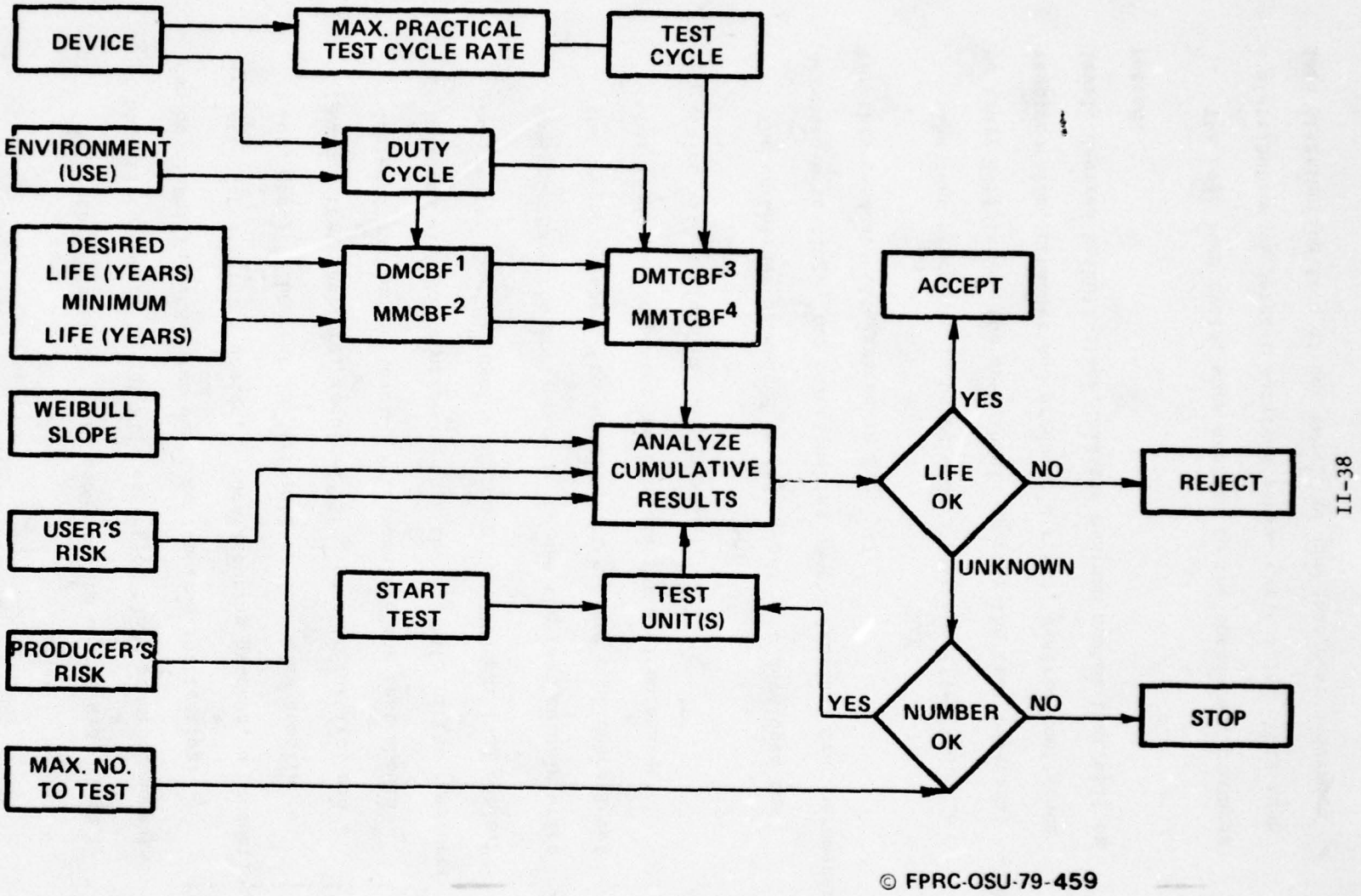
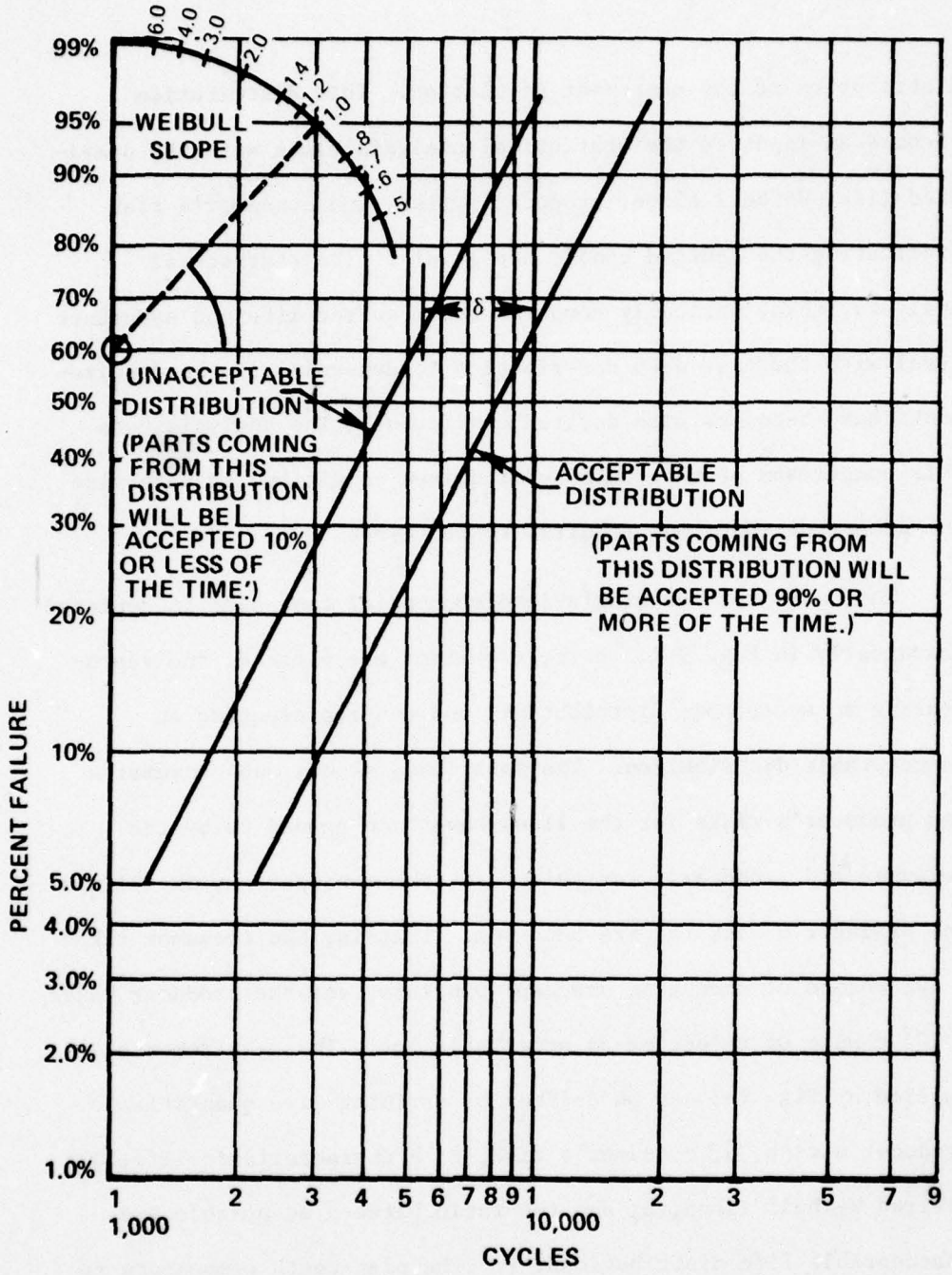


Fig. V-1. Critical Parameters for Sequential Life Tests

distribution of the component population. This distribution becomes an input to the statistical analysis along with the specified life, Weibull slope, producer's risk, and consumer's risk (indicating the desired confidence level). The statistical analysis, then, basically compares the required life and assurance level with the test data distribution to determine if the requirements have been met with desired confidence. The analysis uses this comparison of test data and required conditions to determine the number of specimens required to be tested.

The basis for the single-load sequential test plan is depicted graphically in Fig. V-2. Here, two lines are plotted, one representing an acceptable distribution, and one representing an unacceptable distribution. The locations, slope, and consumer's and producer's risks for the lines have been agreed to by the consumer and producer. For this case, the consumer's risk (β) and the producer's risk (α) are both 10%. That is, the consumer takes a 10% chance of accepting unacceptable lots, and the producer takes a 10% chance of rejecting an acceptable lot. The requirements implied by Fig. V-2 can be defined by defining five quantities: producer's risk, α; consumer's risk, β; a characteristic life, θ_o; desired Weibull slope, b; and the ratio between acceptable and unacceptable life distributions, γ. The plan tests components to failure sequentially until a satisfactory amount of evidence is built up that the actual component life distribution is either better than the acceptable distribution or worse than the unacceptable distribution. An important feature of the plan is that a maximum



Sequential Test Plan Graphic Aid for Understanding.

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Fig. V-2

limit is placed on the number of components that will be tested. Once this number is achieved, the test is terminated and a decision to accept or reject is forced.

Use of the plan involves testing components one at a time. After each test, a quantity known as the Sum of Life Factors is calculated and compared to limits set forth in a sequential test plan table. Notation used with the plan is:

θ_o = characteristic life (desired)

b = Weibull slope (desired)

γ = characteristic life ratio

r = number of items tested

r_o = maximum number to test

α = producer's risk

β = consumer's risk

S_L = satisfactory life factor criterion

U_L = unsatisfactory life factor criterion

The Life Factor is defined as:

$$(\text{Life})^b \qquad \text{(V-1)}$$

The sum of Life Factors is computed after each test by summing Life Factors for all tests completed. The table setting forth for the limits is computed as follows:

STEP 1: Define acceptable values for characteristic life, θ_o, Weibull slope, b, and characteristic life ratio, γ, using a Weibull plot

such as Fig. V-2.

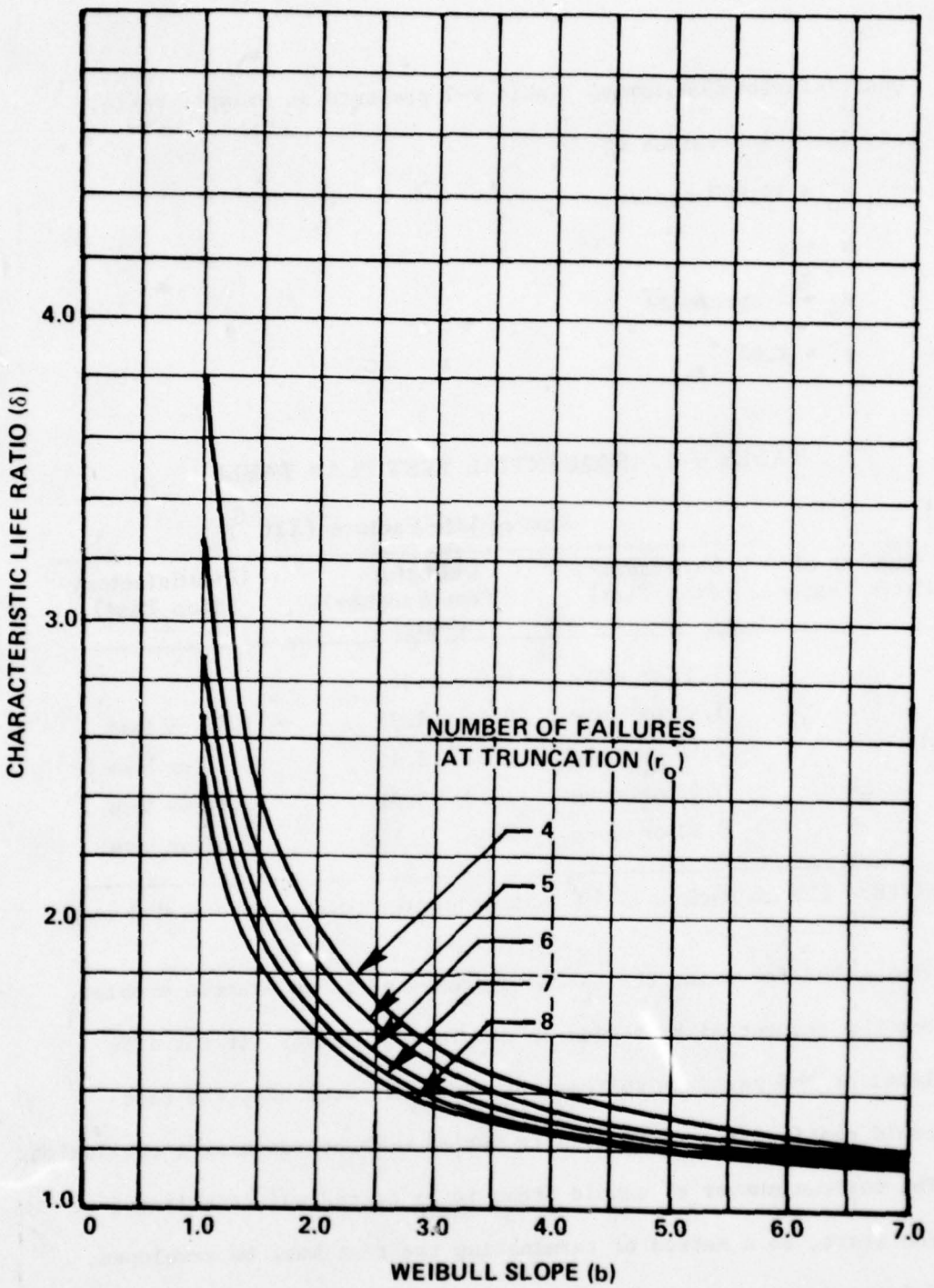
STEP 2: Determine the maximum number of components that will be tested before reaching a decision, r_o. Available test specimens and test facilities, as well as expected test length, must be weighed. The maximum test number, r_o, and the characteristic life ratio, γ, cannot be chosen independently, however. For a large γ, the acceptable life distribution is quite different from the unacceptable distribution, and few test specimens are required to discriminate between them. As γ is decreased bringing the two curves closer together, it is more difficult to discriminate between them and more specimens are required. Figure V-3 gives the minimum number of failures at truncation that should be used for given values of γ and b.

STEP 3: Calculate the values of satisfactory life factor criterion, S_L, and unsatisfactory life factor criterion, U_L, for values of r from one to r_o.

$$S_L = \frac{\theta_o{}^b[rb(\ell_n\gamma) + 2.1972]}{\gamma^b - 1} \qquad \text{(V-2)}$$

$$U_L = \frac{\theta_o{}^b[rb(\ell_n\gamma) - 2.1972]}{\gamma^b - 1} \qquad \text{(V-3)}$$

STEP 4: Prepare the Sequential Test Plan Table as in Table V-1, listing values of r under "Number of Parts Tested," S_L under "Satisfactory," and U_L under "Unsatisfactory." In the "Doubtful" column list the ranges between the values in the "Satisfactory" and



Characteristic Life Ratio vs. Weibull Slope for Various Number of Failures at Truncation.

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Fig. V-3

"Unsatisfactory" columns. Table V-1 presents an example table, calculated for values of

θ_o = 10,000 cycles

b = z

r_o = 5 specimens

γ = 1.82

TABLE V-1. SEQUENTIAL TEST PLAN TABLE

Number of Parts Tested	Sum of Life Factors ($X10^{-8}$) Satisfactory (Stop Test)	Doubtful (Test Another) (Part)	Unsatisfactory (Stop Test)
1	1.47 or more	0.0 - 1.46	
2	1.99 or more	0.10 - 1.98	0.09 or less
3	2.50 or more	0.61 - 2.49	0.60 or less
4	3.02 or more	1.13 - 3.01	1.12 or less
5	3.54 or more	1.65 - 3.53	1.64 or less

NOTE: The coefficient of 10^8 was dropped in each case for simplicity.

The method for using the table is presented in an example problem. Now the sequential test plan is complete. However, if the life level of the parts is neither very good nor very bad, the test could continue for an indefinite period without reaching a conclusion. The maximum number of sample items to be tested was established at the start, so a method of terminating the test must be developed. This termination must also give a yes or no answer to the question of part acceptability. This is accomplished as shown in the following

step.

STEP 5: From Table V-2, select the "Truncation Factor" which corresponds to the value of r_o which was selected in Step 1.

TABLE V-2. TRUNCATION FACTORS

r_o	Truncation Factor
4	1.75
5	2.43
6	3.15
7	3.90
8	4.66

For this example: Truncation Factor = 2.43

Next, determine the value of the Sum of Life Factors at which to truncate the test from the following expression:

$$\text{Sum of Life Factors at Truncation} = \theta^b \times \text{Truncation Factor} \quad \text{(V-4)}$$

For this example: Sum of Life Factors at Truncation = $(10.000)^2 \times 2.43 = 2.43 \times 10^8$

Revise the Sequential Test Plan Table as follows (see Table V-3):

1. Change the last value in the "Unsatisfactory Life" column to read "Less than 2.43."
2. Limit the "Satisfactory Life" values to a maximum of 2.43.
3. Change the values in the "Doubtful Life" column to correspond

with the values in the "Satisfactory Life" and "Unsatisfactory Life" columns.

STEP 6: In some cases it may be desirable to have a minimum life requirement in addition to the TABLE. If a minimum life is desired, a low enough value should be selected so that if a part fails below it, that will be a clear indication of an out-of-control process.

For this example: Figure V- 2 shows that at most only 1% of the parts from an "Acceptable Distribution" are expected to fail by about 1000 cycles. Thus a minimum life requirement of 1000 cycles seems reasonable.

A minimum life requirement will change the TABLE so that an "Unsatisfactory Life" conclusion can be reached at the first level.

This minimum life requirement corresponds to a minimum life factor, for this example, of:

$$(1000)^b = (1000)^2 = 10^6 \qquad \text{(V-5)}$$

This addition is shown on the revised sequential test plan table, Table V-3.

The minimum life requirement also applied to the other parts tested but does not show up in the table. That is, each part tested must exceed the minimum life requirement or else the test will be stopped there and the life considered "Unsatisfactory."

TABLE V-3 (revised). SEQUENTIAL TEST PLAN TABLE

Number of Parts Tested	Sum of Life Factors ($X10^{-8}$) Satisfactory (Stop Test)	Doubtful (Test Another Part)	Unsatisfactory (Stop Test)
1	1.47 or more	0 - 1.46	0.01 or less
2	1.99 or more	.10 - 1.98	.09 or less
3	2.43 or more	.61 - 2.42	.60 or less
4	2.43 or more	1.13 - 2.42	1.12 or less
5	2.43 or more	---	Less than 2.43

Example:

With this test plan, components are randomly selected and then tested to failure one-at-a-time. After each failure the results of the test up to that point are compared with the TEST PLAN TABLE to determine if a conclusion can be reached about the reliability of the lot of parts. This procedure is best illustrated by an example:

Assume that a sample of five components has been selected and the first component tested to failure--which occurs at 5000 cycles. The "Life Factor" is calculated as .25. Comparing this "Life Factor" with Table V-3 above, we see that it falls in the "Doubtful Life" region and another part must be tested.

The second part is tested and fails at 9000 cycles. Its "Life Factor" is .81. The "Sum of Life Factors" for the first two levels is 1.06. When we compare this "Sum" to the second

level in Table V-3, it is still in the "Doubtful Life" region, and another part must be tested.

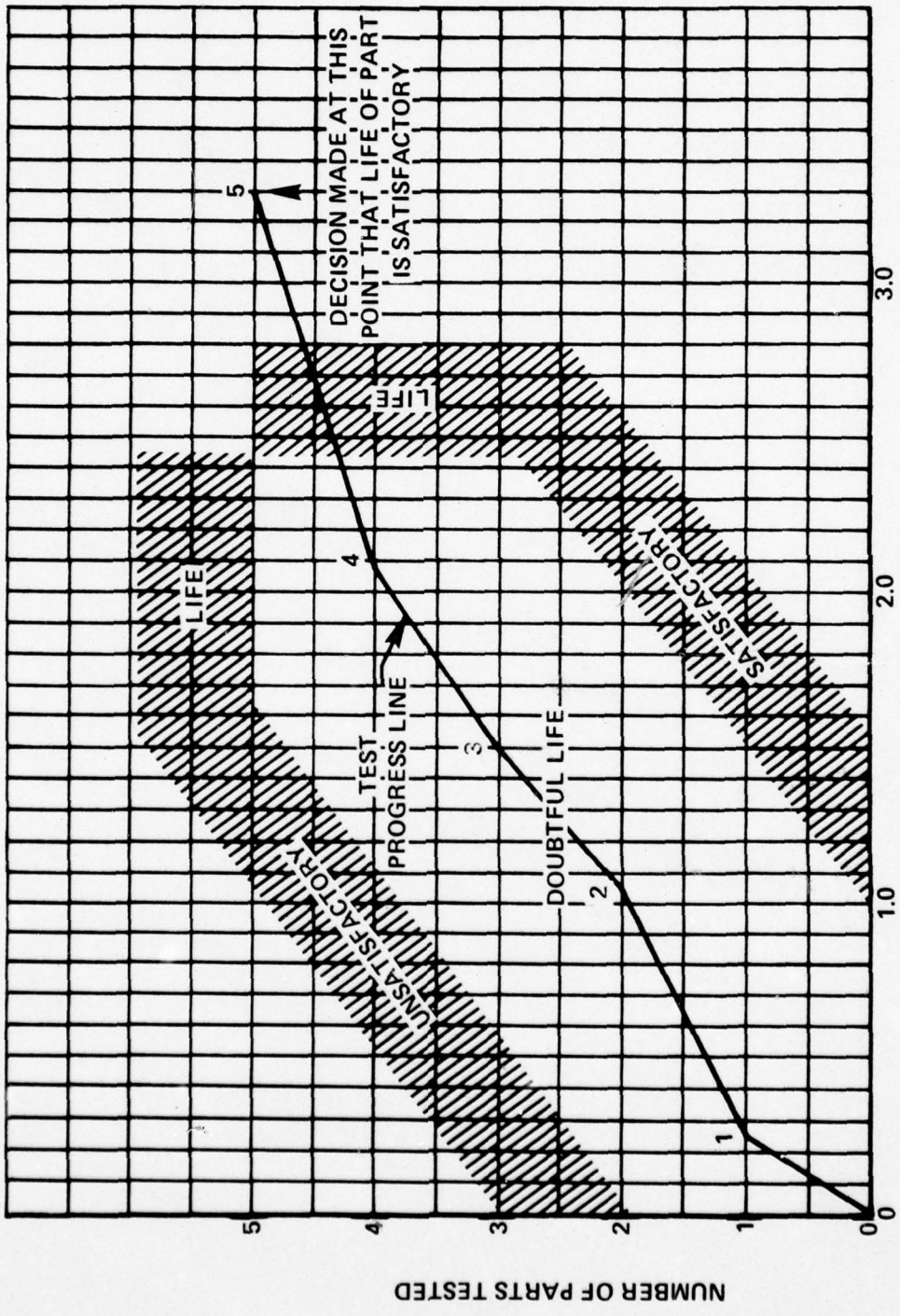
The third part is tested and fails at 6700 cycles. Its "Life Factor" is .45. The "Sum of Life Factors" for the first three levels is 1.52. When we compare this "Sum" to the third level in Table V-3, it is still in the "Doubtful Life" region, and another part must be tested.

The fourth part is tested and fails at 7500 cycles. Its "Life Factor" is .56. The "Sum of Life Factors" for the first four levels is 2.07. When we compare this "Sum" to the fourth level in Table V-3, it is still in the "Doubtful Life" region and another part must be tested.

The fifth part is tested and fails at 11,000 cycles. Its "Life Factor" is 1.21. The "Sum of Life Factors" for all five levels is 3.28. When we compare this "Sum" to the fifth level in Table V-3, it is in the "Satisfactory Life" region. Thus the reliability of the lot of parts is considered satisfactory and the test is stopped.

Had the "Sum of Life Factors" exceeded the value in the "Satisfactory Life" column at an earlier level, the test would have been stopped at that point. Similarily, if the "Sum of Life Factors" had been less than the value in the "Unsatisfactory Life" column at any level, the reliability would have been considered "Unsatisfactory" and the test stopped at that point.

Figure V-4 shows this procedure in graphic form.



Sequential Test Plan.

Fig. V-4

CHAPTER VI

CONCLUSIONS AND RECOMMENDATIONS

CONCLUSIONS

This project investigated numerous factors related to establishing meaningful, cost-effective life test specifications. Certainly it remains apparent that accelerated testing and sequential testing are valid techniques for reducing test costs and time. Sequential test plans are presented in this report for both single and multiple load tests. It is concluded that in the current *"industrial environment"* the single load test plan is preferable, since:

1. It allows optimizing of the sample size.
2. The technique has been more fully developed and is more generally accepted by industry.

The next section recommends a single load sequential test technique similar to that presented in Chapter V. It is recommended that this concept be used in the formulation of test specifications for components. Appendix K presents guidelines for verifying a recommended test method. Compared with available specifications that only require testing one component to failure or suspension, the procedure in the next section has the potential for significantly increasing the survivability of hydraulic systems. The proposed concept for endurance testing should provide a reasonable confidence

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level for a reasonable cost. Following the constraints of the proposed method will also provide the additional benefits of reduced life cycle costs because: 1) data will be available to implement residual life estimates and optimize planned replacement, 2) the proposed procedure will provide adequate discrimination, and 3) data will be available to calculate component reliability in a specified field environment.

RECOMMENDATIONS

There are two approaches that are popular for deciding if the endurance characteristics of a given component or system are satisfactory. The first defines an acceptable life for a given *"field"* load cycle, and proceeds to test for life using the established field duty cycle. The second establishes a realistic endurance test, tests with the standard procedure, and then given the field duty cycle, uses the laboratory life to calculate an expected life for the component in the field.

The first technique requires testing of the component for each application. This occurs because the results of one set of tests will seldom be exactly what is needed for some new field duty cycle.

The second procedure minimizes the amount of testing that is required since there is only one test procedure. The results of this single test procedure can be used for calculating the life of the component in the majority of cases. For those cases where the

field duty cycle is not known, special techniques for evaluating residual life and subsequently obtaining the expected field life are possible.

The second procedure is recommended here because it reduces the total cost of meaningful testing. The procedure is outlined below:

1. Test the component at the following conditions:

a. Fluid MIL-L-2104

b. Pressure, Maximum . . . . . 1.3 Rated Pressure (5%)

c. Waveform, Pressure (load) . . . . Maximum pressure 50% ± 5% of the cycle.
Minimum pressure more than 30% of the cycle.

d. Pressure, Minimum . . . Less than 10% of the maximum pressure

e. Speed (Velocity) . . . . . Rated

f. Temperature . . . . . 80°C (176°F)

2. Test a minimum of three units to failure or an equivalent life of 1,000,000 at a cycle rate of one hertz. Figure VI-1 shows the maximum number of test cycles for a given cycle rate. If the cycle rate is fixed at one hertz (commonly used in the industry), one variable would be eliminated. Failures at cycle rates other than one hertz can be converted to *"equivalent life"* by multiplying the actual cycle life by the life factor found in Fig. VI-2.

3. Using the *"failure"* data, calculate a least squares

regression line using a Weibull model. If the *"best fit"* line yields an estimate that 90% of the population will last longer than 300,000 cycles (at one hertz), accept the component. If not, reject the component for the proposed rated pressure. See Fig. VI-3.

4. Estimate field lives for the component by using the field duty cycles for temperature and pressure to convert the laboratory life to field life.

For future studies the following recommendations are made:

1. Confirm the specific effects of *"time at load"* and *"time off load"* on fatigue lives for various materials so that: a) more accurate field life predictions can be made, and b) data will be available to design more cost effective accelerated test methods.

2. Establish definite correction factors for temperature on influences on fatigue life. More exact knowledge about this critical parameter could lead to more effective accelerated test codes.

3. Prepare a justification for the use of a minimum of three samples and a maximum of five samples for endurance testing.

4. Prepare a *"design guide"* for converting laboratory lives to field lives.

5. Develop specific techniques for determining residual component fatigue lives and optimizing planned replacements.

6. Continue interaction with industry to develop meaningful endurance

test codes and assist in the verification of those procedures.

7. Conduct a Fluid Power System Reliability Conference to assist industry in understanding accelerated testing, field life predictions and the statistical behavior of endurance tests.

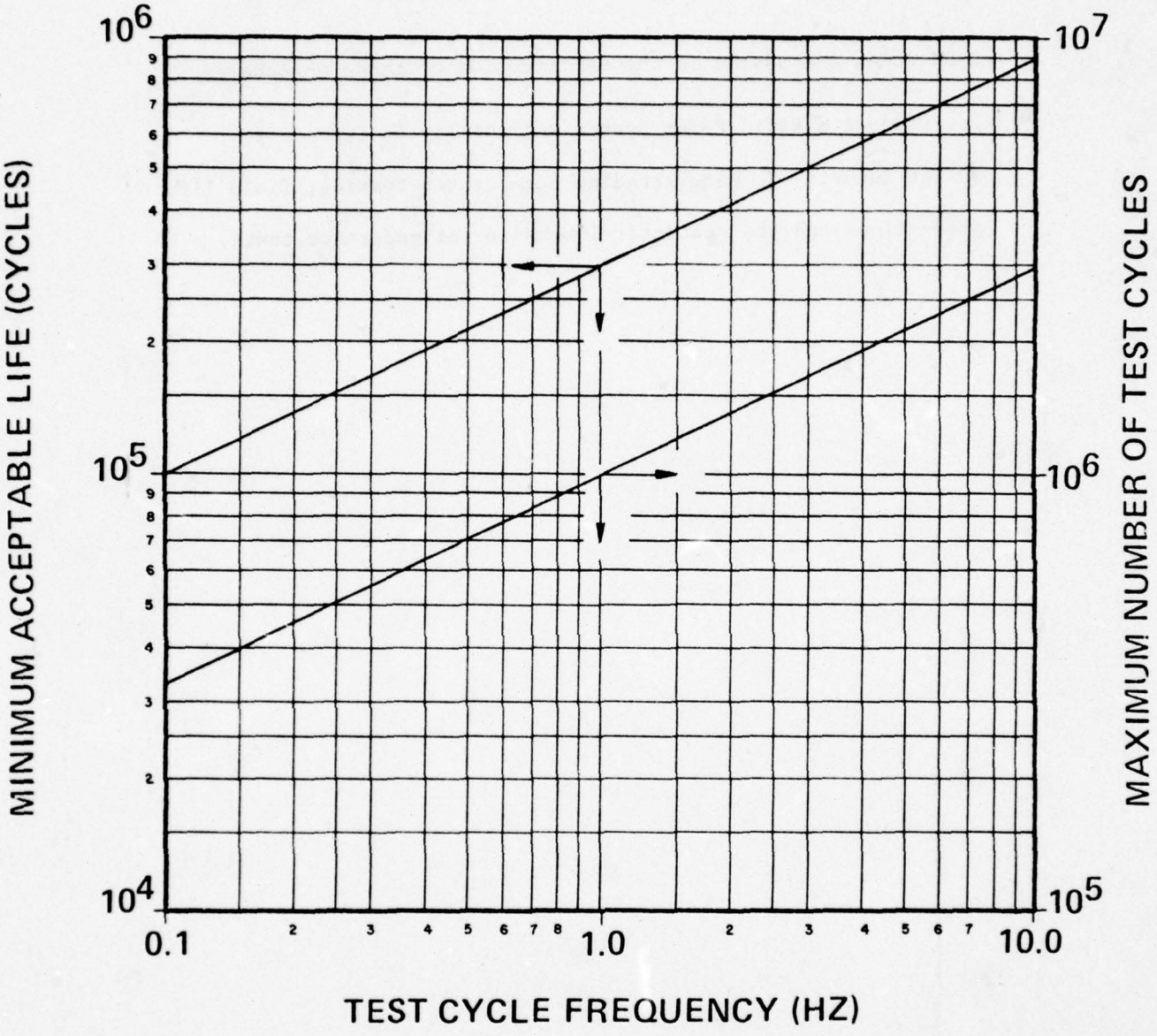


Fig. VI-1. Minimum Acceptable Life & Maximum Number of Test Cycles vs. Test Cycle Frequency

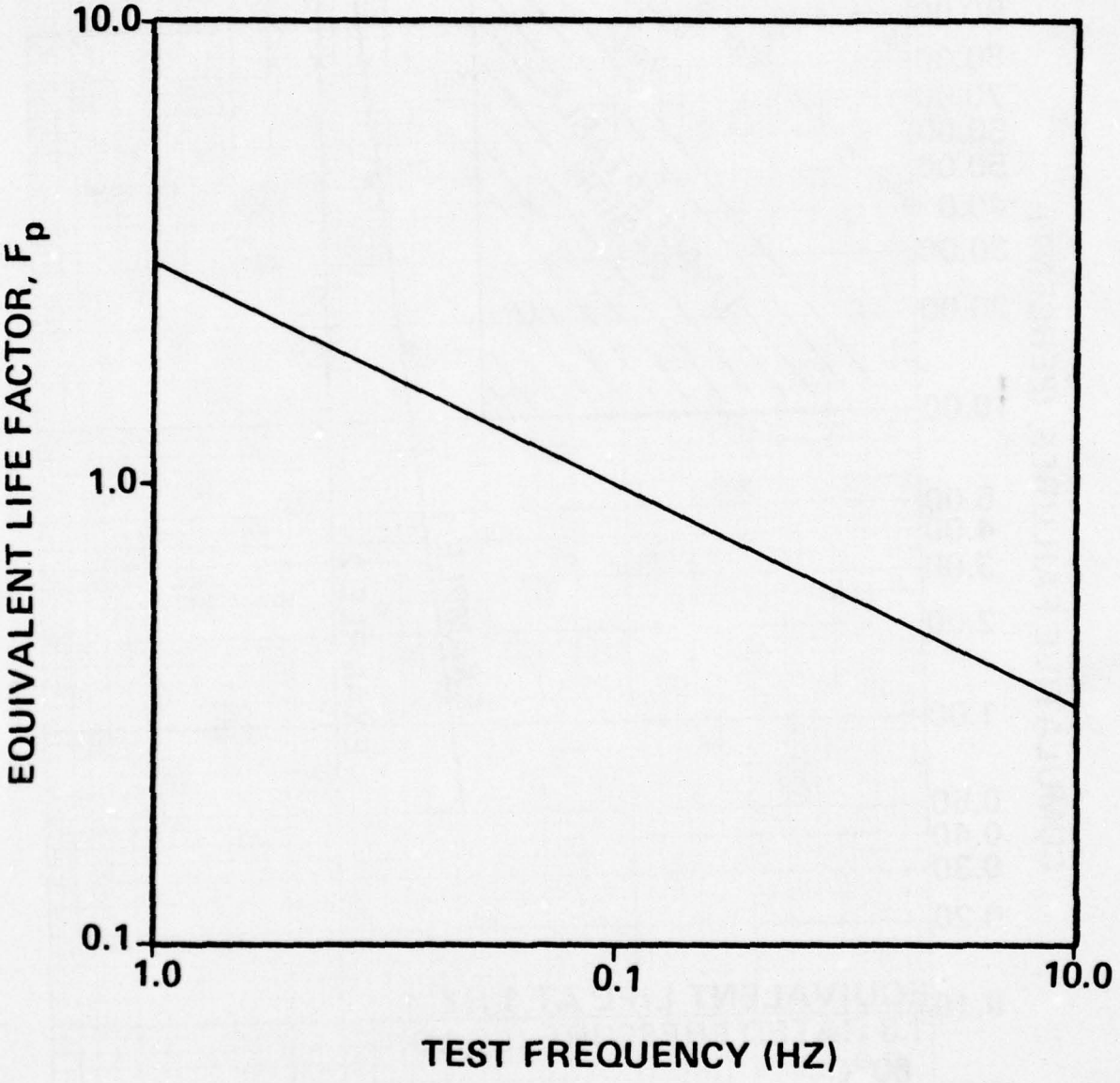


Fig. VI-2. Equivalent Life Factor vs. Test Frequency

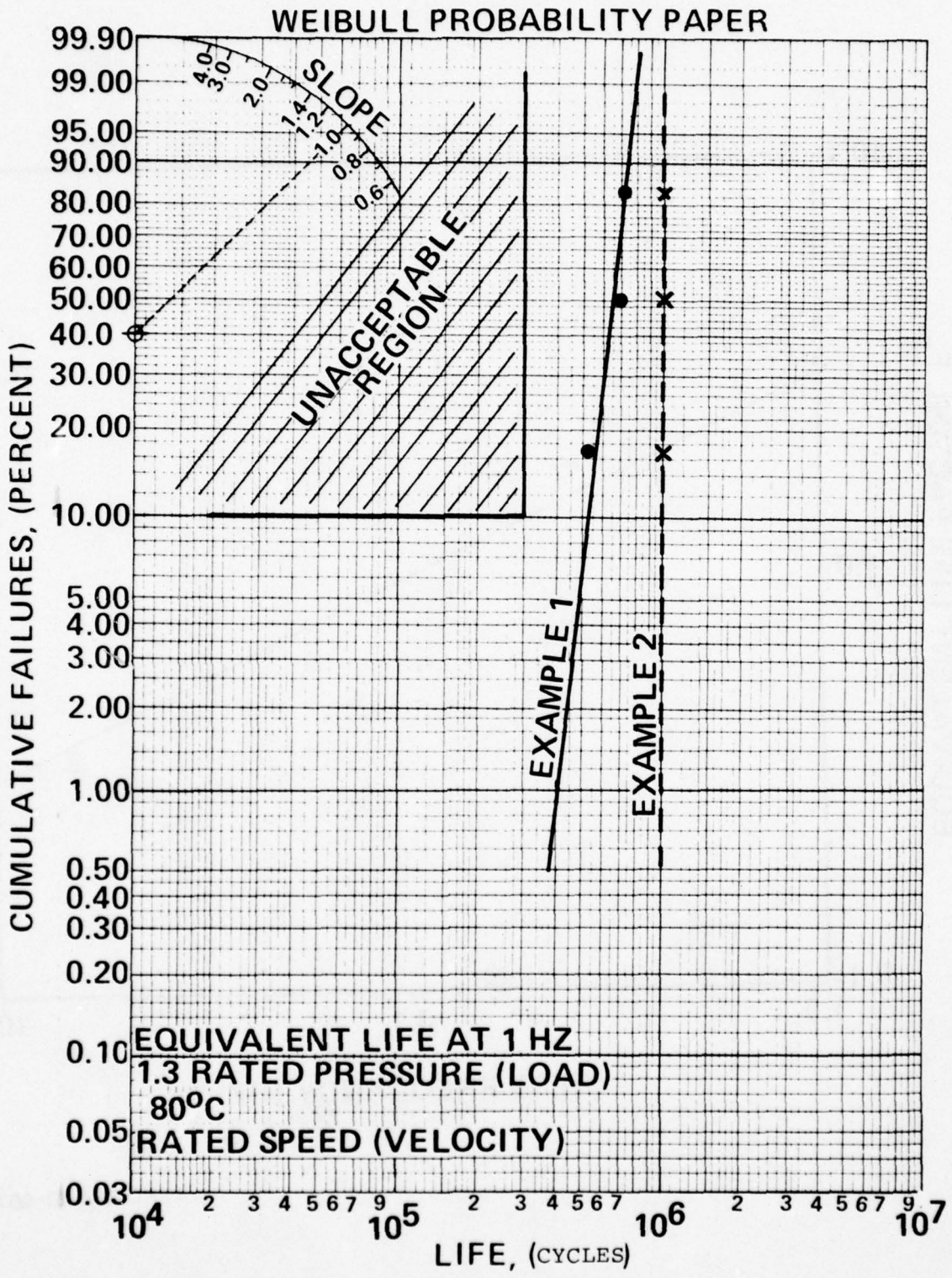


Fig. VI-3. Weibull Probability Plots for Two Examples

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CHAPTER VII

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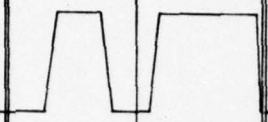
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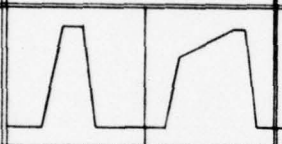
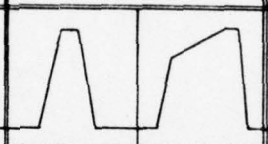
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APPENDIX A

Table A-1. Summary of Pertinent Component and Coupon Failure Data [2].

COMPONENT	PUMP NO. 1		PUMP NO. 2		HOSE NO. 1	
CYCLE FORM						
RISE TIME $\tau_1 = \tau_3$ (Sec)	0.055	0.055	0.025	0.025	0.2	0.095
TIME ON PRESSURE τ_2 (Sec)	0.01	0.46	0.08	0.62	0.03	0.311
P_{max}	P_1	P_1	P_2	P_2	P_3	P_3
AVERAGE CYCLE LIFE FOR SAME MODE OF FAILURE	(2) 2,486,000	(1) 714,000	(3) 318,000	(18) 150,000	(17) 216,000	(16) 122,000

COMPONENT	CYLINDER NO. 1		CYLINDER NO. 2		COUPON SET NO. 1	
CYCLE FORM						
RISE TIME $\tau_1 = \tau_3$ (Sec)	---	---	---	---	0.07	0.07
TIME ON PRESSURE τ_2 (Sec)	Short	Long	Short	Long	0.02	0.46
P_{max}	2500 psi	2250 psi	2500 psi	2250 psi	25 psi	25 psi
AVERAGE CYCLE LIFE FOR SAME MODE OF FAILURE	110,466 (2)	65,644 (5)	109,166 (2)	71,811 (3)	31,878 (2)	20,870 (2)

COMPONENT	COUPON SET NO. 2		COUPON SET NO. 3		COUPON SET NO. 4	
CYCLE FORM						
RISE TIME $\tau_1 = \tau_3$ (Sec.)	0.05	0.05	0.05	0.05	0.05	0.05
TIME ON PRESSURE τ_2 (Sec.)	0.15	1.50	0.15	1.50	0.15	1.50
P_{max}	21 psi	21 psi	16 psi	16 psi	15 psi	15 psi
AVERAGE CYCLE LIFE FOR SAME MODE OF FAILURE	26,734 (5)	28,053 (5)	116,578 (5)	50,381 (5)	257,888 (4)	71,239 (3)

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APPENDIX B

Table B-1. Data Obtained from Radial Bending Tests on 2024-T351 Coupons. "Time at load" equals 1.50 sec. for long cycle and 0.15 sec. for short cycle. "Time off load" equals 1.50 sec. for both cases [2]. Mean stress, S_m, equals alternating stress, S_a.

TEST	SHORT "TIME ON LOAD"		LONG "TIME ON LOAD"	
PRESSURE	Coupon Number	Cycle Life at "Long Time on Load"	Coupon Number	Cycle Life
(0.014 daN/mm^2)	2	28,633	1	41,316
	4	35,148	3	37,490
21 psi	6	26,675	5	20,035
Coupon Set No. 2	8	24,116	7	17,948
	10	19,397	9	23,480
S_a = 29.9 ksi				
(20.6 daN/mm^2	Mean	26,799	Mean	28,053
19 psi	11	37,999	12	38,843
(0.011 daN/mm^2)	13	146,820	14	89,820
	15	183,270	16	36,544
16 psi	17	74,282	18	40,296
Coupon Set No. 3	19	72,837	20	43,700
	21	124,588	22	54,811
S_a = 22.8 ksi	23	97,671	24	37,115
(15.7 daN/mm^2)	Mean	116,578	Mean	50,381
(0.01 daN/mm^2)	26	121,168	27	75,148
15 psi	28	95,999	30	78,883
Coupon Set No. 4	33	733,830	34	59,686
	35	80,556	36	84,955
S_a = 21.4 ksi	Mean	257,888		74,418
14 psi	---	------	25	1,576,632

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APPENDIX C

EQUIVALENT TIME AT FULL LOAD

The concept of using a single quantity, the *"equivalent time at full load,"* T_e, to describe the test cycle allows a helpful simplification of the evaluation of Eq. (II-2). (See Ref [3] for a complete development of the concept and the associated expressions.) The *"equivalent time at full load,"* T_e, is defined as the amount of *"time at load"* at a constant load equal to the maximum load occurring during the load cycle, P_{max}, which would produce the same amount of strain as that produced by the test cycle. That is, the actual test cycle is replaced by an equivalent cycle having a constant load of P_{max} for time T_e. An expression to evaluate this quantity for any given segment of a load cycle has been developed by setting the integral of Eq.(II-2) for a generalized load cycle equal to the integral for the equivalent cycle. For any load cycle segment which can be expressed by:

$$P(t) = P_i + \frac{P_i + 1}{\tau_i}(t - t_i) \qquad \text{(C-1)}$$

an expression for the *"equivalent time at load"* can be derived as:

$$t_{ei} + 1 = \left\{ \frac{P_i}{P_m} (t_{ei} + \tau_i)^{1-n} + (1 - \frac{P_i}{P_m}) t_{ei}^{1-n} + \frac{P_i + 1^{-P_i}}{P_m \ \tau_i} \left[(\frac{1 - n}{2 - n})(t_{ei} + \tau_i)^{2-n} + \frac{t_{ei}^{2-n}}{(2-n)} - t_{ei} (t_{ei} + \tau_i)^{1-n} \right] \right\}^{\frac{1}{1-n}}$$

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This *"equivalent time"* t_{ei+1} can now be directly solved for, since all other parameters are known from the segment load cycle or from analysis of the preceeding segment.

The test cycle segments are thus analyzed sequentially, using this expression. Evaluation of the last segment yields the *"equivalent time at load"* for the entire test cycle, T_e.

Table D-1. Summary of Verifications of Fatigue Life Prediction Technique [3]. "Time off pressure" greater than or equal to "time on pressure." Material parameters used were n = 0.5, c = 1.05.

COMPONENT	PUMP NO. 1		PUMP NO. 2		HOSE NO. 1		COUPON SET A*	
CYCLE FORM								
RISE TIME $\tau_1 = \tau_3$	0.055 sec.	0.055 sec.	0.025 sec.	0.025 sec.	0.2 sec.	0.095 sec.	0.01 sec.	0.01 sec.
TIME ON PRESSURE τ_2	0.01 sec.	0.46 sec.	0.08 sec.	0.62 sec.	0.03 sec.	0.311 sec.	0.15 sec	1.50 sec.
P_{max}	P_1	P_1	P_2	P_2	P_3	P_3	P_4*	P_4*
ACTUAL MEAN CYCLE LIFE FOR SAME MODE OF FAILURE	2,486,000 (2)	714,000 (1)	318,000 (3)	150,000 (18)	216,000 (17)	122,000 (16)	173,105 (10)	59,996 (10)
PREDICTED MEAN LIFE	----	769,000	----	144,000	----	133,000	----	58,700
ERROR	8%		4%		9%		+2%	

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*Coupon Set A consists of the 15 psi (0.01 daN/mm^2) and 16 psi (0.011 daN/mm^2) data of Appendix B.

APPENDIX E

LIFE PREDICTION COMPUTER PROGRAM

The material failure number prediction technique developed in the Technical Background section is performed by the computer program "Flolife." The program is stored on TSO at Oklahoma State University under the user number "U15454B" having the password "FPRC 11." The program predicts the field endurance life (N_2) given the lab endurance life, certain material constants, and coordinates describing the pressure-time characteristics of the lab and field waveforms.

The program was run to predict the field endurance life (with the field duty cycle of Fig. E-1) corresponding to a laboratory test life of one million cycles using the test cycle of Fig. E-2.

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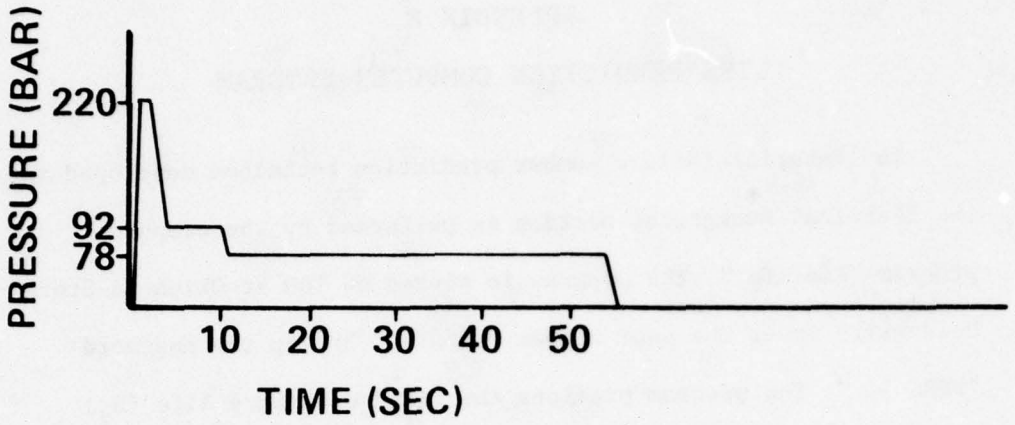


Fig. E-1. Field Duty Cycle for Example Problem.

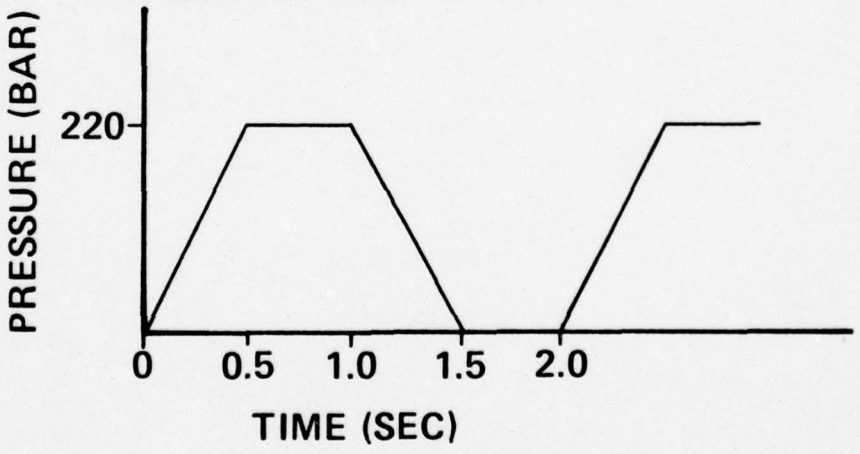


Fig. E-2. Laboratory Test Cycle for Example Problem. FPRC OSU 79-439

```
run fidlife.fort
GL COMPILER ENTERED
SOURCE ANALYZED
PROGRAM NAME = MAIN
*  NO DIAGNOSTICS GENERATED
SOURCE ANALYZED
PROGRAM NAME = TIMEQ
*  NO DIAGNOSTICS GENERATED
    *STATISTICS*  NO DIAGNOSTICS THIS STEP :


INPUT DUCTILITY COEFF & DYNAMIC STRAIN EXP (2F10.3)

     1.050      0.5
INPUT NO. OF CYCLES TO FAILURE FROM LAB TEST (E10.2)
   1.00e06
INPUT NO. OF PTS. ON P-T CURVE FOR LAB LOAD CYCLE (I10)
         4
INPUT P-T COORDINATES FOR THE LAB LOAD CYCLE (2F10.3)

        P          T
      0.          0.
    220.          0.500
    220.          1.
      0.          1.500
NO. OF POINTS ON P-T CURVE FOR FIELD LOAD CYCLE (I4)
        8
INPUT P-T COORDINATES FOR THE FIELD LOAD CYCLE (2F10.3)

        P          T
      0.          0.
    220.          0.500
    220.          1.500

     92.          1.790
     92.         10.500
     78.         10.790
     78.         55.290
      0.         55.790
END OF DATA
FIELD ENDURANCE LIFE=0.25370406CYCLES
READY
```

```
C
C       THIS PROGRAM PREDICTS THE ENDURANCE LIFE OF A
C       COMPONENT HAVING A KNOWN FIELD DUTY CYCLE FROM
C       THE LAB TEST LIFE AT A KNOWN DUTY CYCLE.
C
        IMPLICIT REAL*8(A-H,O-Z)
        DIMENSION P(20),T(20),TE(2),TTE(20),TAU(20),SS(20)
  1     FORMAT(///,' INPUT DUCTILITY COEFF & DYNAMIC STRAIN EXP (2F10.3)')
  2     FORMAT(' INPUT NO. OF CYCLES TO FAILURE FROM LAB TEST (E10.2)')
  3     FORMAT(' INPUT NO. OF PTS. ON P-T CURVE FOR LAB LOAD CYCLE (I10)')
  4     FORMAT(' INPUT P-T COORDINATES FOR THE LAB LOAD CYCLE (2F10.3)')
  5     FORMAT(' NO. OF POINTS ON P-T CURVE FOR FIELD LOAD CYCLE (I10)')
  6     FORMAT(' INPUT P-T COORDINATES FOR THE FIELD LOAD CYCLE (2F10.3)')
 111    FORMAT(' FIELD ENDURANCE LIFE=',E10.4,'CYCLES')
  7     FORMAT(' END OF DATA')
  8     FORMAT(/,'          P          T  ')
 104    FORMAT(2F10.3)
 102    FORMAT(I10)
 101    FORMAT(E10.2)
 100    FORMAT(2F10.3)
        PRINT 1
        READ 100,C,XN
        PRINT 2
        READ 101,XN1
        PRINT 3
        READ 102,K
        J=1
        PRINT 4
        PRINT 8
        DO 103 I=1,K
 103    READ 104,P(I),T(I)
        CALL TIMED(P,T,K,XN,C,TE,J)
        PRINT 5
        READ 102,K
        J=11
        PRINT 6
        PRINT 8
        DO 105 I=1,K
 105    READ 104,P(I),T(I)
        PRINT 7
        CALL TIMED(P,T,K,XN,C,TE,J)
        S=(1-XN)/C
        RL=(TE(1)/TE(2))**S
        XN2=XN1*RL
        PRINT 111,XN2
        STOP
        END
```

```
      SUBROUTINE TIMEO(P,T,K,XN,C,TE,J)
      IMPLICIT REAL*8(A-H,O-Z)
      DIMENSION P(20),T(20),TE(2),TTE(20),TAU(20),SS(20)
      DO 99 I=1,K
   99 SS(I)=P(I)
      DO 101 I=2,K
      IF(P(1)-P(I))100,101,101
  100 PP=P(1)
      P(1)=P(I)
      P(I)=PP
  101 CONTINUE
      KP1=K+1
      KM1=K-1
      PM=P(1)
      DO 102 I=1,KM1
      IP1=I+1
  102 TAU(I)=T(IP1)-T(1)
      DO 110 I=1,K
  110 P(I)=SS(I)
      TTE(1)=0.
      W=1/(1-XN)
      WP1=2-XN
      S=1/(W*WP1)
      Z=1-XN
      DO 103 I=1,KM1
      IP1=I+1
      R=P(I)/PM
      SUM=(TTE(I)+TAU(I))**Z
      SUM1=(TTE(I)+TAU(I))**WP1
      SUM2=(TTE(I))**WP1
      SUM3=(P(IP1)-P(I))/(PM*TAU(I))
      TTE(IP1)=(R*SUM+(1-R)*(TTE(I)**Z)+SUM3*(S*SUM1+SUM2/WP1-TTE(I)*SUM
     1 ))**W
  103 CONTINUE
      TE(J)=TTE(K)
      RETURN
      END
```

APPENDIX F

STATISTICAL TABLES

NUMBER OBSERVED	SIZE OF SAMPLE, n						FRACTION OBSERVED	SIZE OF SAMPLE	
f	10	15	20	30	50	100	f/n	250	1000
0	0 27	0 20	0 15	0 10	0 07	0 4	0.00	0 1	0 0
1	0 40	0 31	0 23	0 17	0 11	0 5	.01	0 4	0 2
2	3 61	2 37	1 30	1 21	0 14	0 7	.02	1 5	1 3
3	8 62	5 45	4 36	2 25	1 17	1 8	.03	1 6	2 4
4	15 74	9 56	7 42	4 30	2 19	1 10	.04	2 7	3 5
5	22 78	14 64	10 47	6 33	3 22	2 11	.05	3 9	4 7

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Table F-1. 95% Confidence Interval (Percent) for Binomial Distribution [2].

DEGREES OF FREEDOM	PROBABILITY OF A GREATER VALUE						
	0.950	0.900	0.750	0.500	0.250	0.100	0.050
1	....	0.02	0.10	0.45	1.32	2.71	3.84
2	0.10	0.21	0.58	1.39	2.77	4.61	5.99
3	0.35	0.58	1.21	2.37	4.11	6.25	7.81
4	0.71	1.06	1.92	3.36	5.39	7.78	9.49
5	1.15	1.61	2.67	4.35	6.63	9.24	11.07

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Table F-2. Cumulative Distribution of Chi-Square [2].

DENOMINATOR d.f.	PROBABILITY OF A LARGER F				
	0.250	0.100	0.050	0.010	0.005
1	7.50	49.5	199.5	5403	20,000
2	3.00	9.00	19.00	99.00	199.0
3	2.28	5.46	9.55	30.82	49.80
4	2.00	4.32	6.94	18.06	26.28
5	1.85	3.78	5.79	13.27	18.31
6	1.76	3.46	5.14	10.92	14.59

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Table F-3. Values of F for Numerator Degrees of Freedom Equals Two [2].

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APPENDIX G

RESPONSE CURVE COMPUTATIONS FOR THREE SPECIMENS AT EACH LOAD

Table G-1. Computations for Fitting a Response Curve by Method of Least Square Fitted Values.

APPLIED STRESS	CODED VALUE, x	PERCENT SURVIVAL, p	TRANSFORMED VALUES, y	x^2	xy	y_f	p_f
4000	1	1.00	-3.0	1	-3.0	-2.12	
4150	2	.67	-.42	4	-.84	-1.36	
4300	3	.67	-.42	9	-1.26	-.6	
4450	4	.33	+.42	16	+1.68	.16	
4600	5	.33	+.42	25	+2.1	.92	
SUM	15		-3.0	55			
	x = 3		y = -.6				

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$$y_f = a + b\,(x-\bar{x}) \qquad \text{(G-1)}$$

$$a = \bar{y} \qquad \text{(G-2)}$$

$$b = \frac{\sum(xy) - k\bar{x}\,\bar{y}}{\sum x^2 - k\bar{x}^2} \qquad \text{(G-3)}$$

$$y_f = -0.6 + 0.76\,(x-\bar{x}) \qquad \text{(G-4)}$$

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APPENDIX H

RESPONSE CURVE COMPUTATIONS FOR FOUR SPECIMENS AT EACH LOAD

Table H-1. Test Data for Example with Four Specimens Tested at Each Pressure.

PRESSURE	NO. OF PUMPS TESTED	NO. SURVIVING	PERCENT SURVIVAL
400	4	4	1.00
415	4	3	0.75
430	4	3	0.75
445	4	2	0.50
460	4	1	0.25

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Table H-2. Information for Least Squares Fit of Response Curve.

APPLIED STRESS	CODED VALUE, x	PERCENT SURVIVAL, p	TRANSFORMED VALUE, y	x^2	xy	y_f
400	1	1.00	-3.0	1	-3.0	-2.33
415	2	.75	-.67	2	-1.34	-1.53
430	3	.75	-.67	9	-2.01	-.73
445	4	.50	0	16	0	.067
460	5	.25	.67	25	3.35	.868
	SUM 15		y = -0.734	55	-3	
	x = 3					

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Table H-3. Information for Determination of Confidence Intervals.

APPLIED STRESS (BARS)	CODE NUMBER	$(x - \bar{x})^2$	$A = \frac{(x - \bar{x})^2}{\Sigma (x - \bar{x})^2}$	$B = (1/k + A)^{1/2}$	$Bs\,(2F_{2,3})^{1/2}$	y_f	y VALUES		p VALUES	
							UPPER	LOWER	UPPER	LOWER
400	1	4	.40	.77	1.68	-2.34	-0.66	-4.02	74.5%	100.0%
415	2	1	.10	.55	1.20	-1.54	-0.34	-2.74	63.3%	99.69%
430	3	0	0.0	.45	0.98	-0.734	0.25	-1.71	49%	95.6%
445	4	1	.10	.55	1.20	0.067	1.27	-1.13	10.2%	87.1%
460	5	4	.40	.77	1.68	0.868	2.55	-0.81	0.54%	79.1%

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$$b = \frac{\sum xy - k\bar{x}\,\bar{y}}{\sum x^2 - k\bar{x}^2} \qquad \text{(H-1)}$$

$$a = -0.734 \qquad \text{(H-2)}$$

$$y_f = -0.734 + 0.801(x-\bar{x}) \qquad \text{(H-3)}$$

$$s = \left[\sum \frac{(y-y_f)^2}{k-2}\right]^{\frac{1}{2}} = 0.697 \qquad \text{(H-4)}$$

$$F(2,3) = 5.46 \qquad \text{(H-5)}$$

APPENDIX I

THE WEIBULL DISTRIBUTION

In order to statistically analyze the test data, a failure distribution model is desired. Analysis of the data is possible without assuming a distribution form, but large sample sizes are required to attain reasonable confidence levels. Since we are concerned with analyzing the test lives at a single load level, it is desired to use the distribution of failures versus life at the given load (rather than the distribution of failures versus load at a given life). For fatigue failures, the normal distribution has not been found to be a useful model. What is really needed is a distribution function that can change not only its location and density, but also its shape to fit the actual test data. The Weibull distribution is such a distribution. This is important because the same basic distribution form can be used to model components having failure rates that are constant (chance failures), decreasing (infant mortality), and increasing (wear or fatigue) [14].

The expression for the Weibull distribution function is:

$$F(x) = 1 - e^{-\left(\frac{x}{\theta}\right)^{\beta}} \qquad \text{(I-1)}$$

where β is the shape of the parameter which allows the unique flexibility of the distribution. A β value of less than one implies a decreasing failure rate, β equal to one implies a

constant failure rate, and β greater than one implies an increasing failure rate. The location of the distribution is given by θ, the location parameter.

The thing that makes the Weibull distribution function so useful for failure analysis is the fact that Weibull Probability Paper has been developed which yields a straight line for any Weibull distribution. The relationship between the axes on the paper is determined by taking the natural logarithm of equation (1) twice to yield:

$$\ln\left(\ln \frac{1}{1-F(x)}\right) = \beta \ln x - \beta \ln\theta \qquad \text{(I-2)}$$

This expression is in the basic form of a straight line when plotted on log log versus log scales. This is the basic form of Weibull Probability Paper. Figure I-2 presents a failure distribution plotted on Weibull Probability Paper as percent failure versus cycle life. The straight line described by the data points indicates that the Weibull distribution is a good model for this data. Once this line is determined, it can be used to provide a best estimate of the cycle life at which any value of percent failure occurs. For instance, the median fatigue life (B_{50}) under these conditions, that is, the number of cycles where half of the components survive and half have failed, is determined to be 175 cycles. The B_{10} life (90% survival) is estimated to be 35 cycles. The parameters θ and β can be estimated directly from

the plot. The shape parameter β can be determined directly from the slope of the line. The slope can be readily determined by drawing a line parallel to the data line on the scale in the upper left corner of the paper. For this case, β is estimated to be 1.2 indicating an increasing failure rate. It can be shown that the percent failed at θ=1 is 63.2% for any value of β. Therefore, the value of the location parameter, θ, can be determined by noting the life where the percent failure is 63.2%. For this case, θ is approximately 230 cycles.

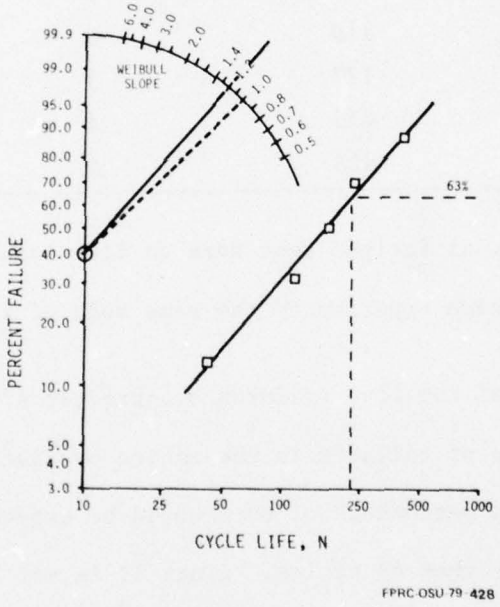


Figure I-2. Weibull Plot of Hydraulic Hose Fatigue Test Data of Table I. Five Hoses were Tested and Failed.

PLOTTING THE DATA

The data are plotted as percent failure (cumulative) versus

cycle life. The first step in plotting the data is to arrange the failures in order with respect to cycle life. It is then necessary to determine what value of percent failure to assign to each data point. The data used in plotting Figure I-2 is listed in Table I-1, arranged in order of increasing cycle life.

TABLE I-1. Information Required for Weibull Analysis of Hose Fatigue Tests.
Five hoses were tested to failure.

Specimen Number	Cycles	Rank	Median Rank
1	42	1	12.9%
2	119	2	31.5%
3	177	3	50.0%
4	235	4	68.6%
5	428	5	87.0%

This data is actual fatigue test data on five identical hydraulic hoses, all of which experienced the same mode of failure.

The first of the five failures occurred at 42 cycles. Considering the distribution of failures in the entire population of those hoses, a certain percentage of them would be expected to fail upon undergoing fewer than 42 cycles. Since it is not known what this percentage actually is, a method of estimating it is desired. This method is furnished by order statistics. Order statistics permit us to make a best estimate of the percent of the hose population failing before 42 cycles, based upon the knowledge that one out of five test specimens failed on or before 42 cycles. This is accomplished

with the use of median rank tables, developed to insure that these estimates are unbiased. Table I-4 presents a median rank table for sample sizes one to twenty. Referring to this table for a sample size of five specimens (hoses), the median rank for the first of the five failures is 0.1294. Thus, the best estimate from this data is the 12.94% of the hose population would fail at or before 42 cycles if subjected to the same test load cycle. Median ranks for the other four data points are found similarly and are tabulated in Table I-2. The median rank for the third point is 50%. This value agrees with our intuitive expectations, since two hoses failed before this point and two failed after it.

These points are then plotted on the Weibull Probability Paper, (Figure I-2) with the number of cycles to failure on the horizontal axis and the percentage point obtained from the median rank on the vertical axis as the percent failure. As noted earlier, the points lie quite closely to the straight line, which was *"eyeballed"* on the graph. If this is not the case, a very careful look at the data is in order. Often, such a plot will indicate that one or more of the components underwent a different mode of failure different from the other components. Weibull plots are very useful for determining *"odd"* modes of failure in this manner. Another situation in which the data will form a curved line results from an offset in the Weibull origin. This can occur if a test has been performed on the specimens prior to the test in question, or if for some reason, no failures will ever occur until the specimens

had undergone a certain number of cycles. If careful examination of the physical evidence reveals no sign of mixed failure modes, and if an offset of the origin is suspected, reference [14] presents mathematical techniques of evaluating this offset.

USING SUSPENDED TEST DATA

Quite often in fatigue testing, not all specimens are failed. Tests of some specimens may be suspended without failure due to equipment malfunctions, scheduling limitations, or undesired modes of failure. Rather than discarding such information, it can be used to improve the estimate of the life distribution. For example, the hose fatigue test previously discussed included three suspended tests not included in Table I-1. Table I-2 presents test results for all eight hoses. Since the first suspended test survives for more cycles than the first four specimens, the ranks for these specimens remains unchanged. However, instead of specimen number one being the first failure out of a five hose sample, it becomes the first failure out of eight hoses, since the other seven hoses all had the opportunity to fail at 42 cycles and did not. The median rank is thus 8.3% instead of 12.9% as before. Median ranks for specimens two, three, and four are determined in the same manner. To obtain a rank for specimen number five, we must determine the increment between specimens four and five resulting from the two suspended specimens. The increment is determined from the formula [14]:

$$I = \frac{(n+1) - (\text{Previous Rank Order})}{1 + (\text{Number of items beyond present suspended set})} \qquad \text{(I-3)}$$

For specimen five the increment is:

$$I = \frac{(8+1) - 4}{1+2} = 1.64 \qquad \text{(I-4)}$$

TABLE I-2. Information required for Weibull Analysis of Hose Fatigue Tests. Five hoses were tested to failure, and tests on three hoses were suspended without failure.

Specimen Number	Cycles	Survive/Fail	Rank	Median Rank
1	42	F	1	8.3%
2	119	F	2	20.2%
3	177	F	3	32.1%
4	235	F	4	44.0%
6	309	S	-	---
7	386	S	-	---
5	428	F	5.67	63.9%
8	428	S	-	---

Thus the rank of specimen five becomes 5.67 which yields a median rank of 63.9%. If the data had contained other failed specimens directly following specimen five, the increment of 1.67 would be used until another suspended specimen was encountered. Then a new increment would be calculated using equation (I-3) [18].

The data of Table I-2 is plotted in Figure I-3.

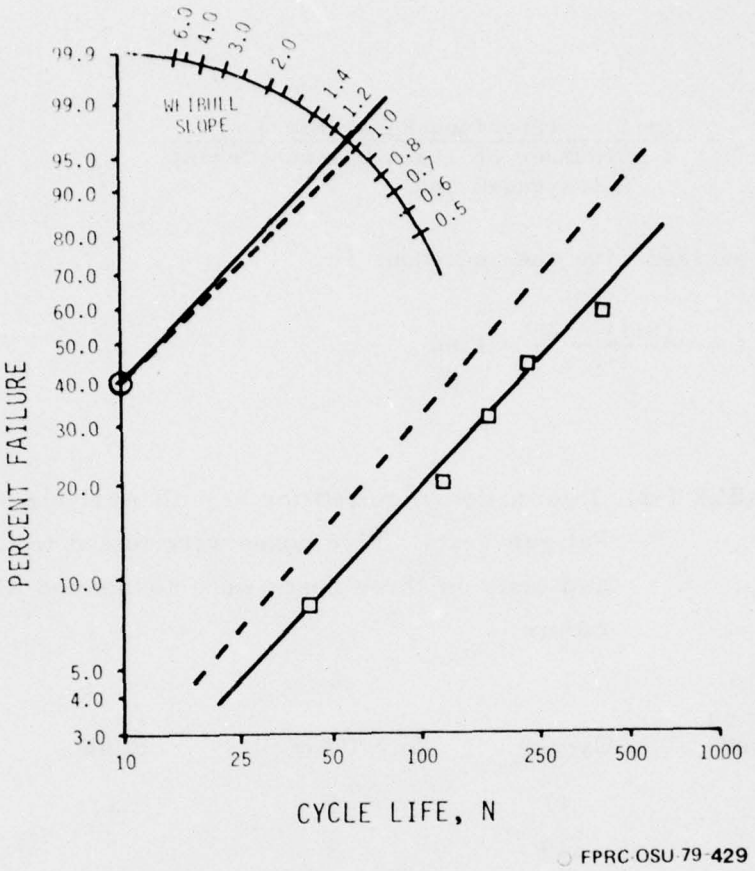


Figure I-3. Weibull Plot of Hydraulic Hose Fatigue Test Data of Table I-2. Eight hoses were tested, but only five were failed. The dashed line indicates the Weibull plot obtained for the five failures without considering the three suspended tests.

The points still describe a straight line. The dashed line in Figure I-3 is the plot obtained in the previous example, which considered the same five failures, but ignored the three suspended tests. As expected, there is considerable difference between the two lines. This underscores the importance of analyzing all usable information to arrive at the best possible estimate of the desired parameters.

CONFIDENCE INTERVALS

The values determined from the median rank tables and plotted previously are unbiased estimates of the life distribution under test. This means that the estimate has a 50% chance of being either too low or too high. In the same vein, the best estimate line can be referred to as the 50% confidence line, since one can be 50% confident that the percent survived will be at least the value predicted by the line.

In addition to this line representing 50% statistical confidence, confidence intervals corresponding to other desired confidence levels can be computed. Figure I-4 illustrates a 90% confidence interval for the data of Table I-2.

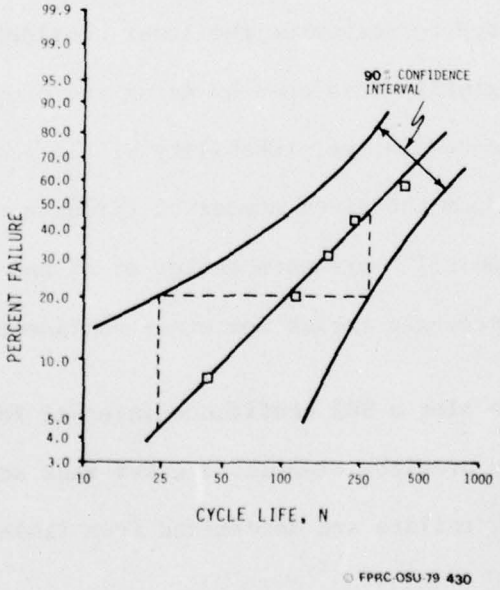


Figure I-4. Best Estimate Line and 90% Confidence Interval for Hydraulic Hose Fatigue Test Data of Table I-3.

We are 90% confident that the true population life distribution lies within the interval. Alternatively, if the test were repeated numerous times upon specimens from the same population, we would expect that the points on the population life distribution would lie within the interval for 90% of the tests.

Computation of the confidence intervals for Weibull plots is a simple procedure, similar to the determination of the plotting points for the *"best estimate"* line. Plotting points for the confidence intervals are obtained in much the same way, except that instead of using a table to determine the median rank, a table of ranks corresponding to the desired confidence is used. For instance, to determine a 90% confidence interval, a table of 95% ranks is consulted to determine the upper confidence limit, and a 5% rank table is consulted to determine the lower confidence limits. These tables were originally developed by using the bionmial distribution to determine the cumulative probability of all possible combinations which would produce the given number of failures with the given sample size. Table I-5 presents tables of 5% and 95% ranks. Reference [14] presents tables for other confidence levels.

In order to plot a 90% confidence interval for the distribution estimated in the previous example, the 95% rank and 5% rank corresponding to each failure are determined from Table I-5 and tabulated in Table I-3.

TABLE I-3. Information Required for Construction of 90% Confidence Interval on Weibull Analysis of Data of Table 2.

Specimen Number	Cycles	Survive/Fail	Rank	95% Rank	5% Rank
1	42	F	1	31.3%	0.7%
2	119	F	2	47.1%	4.7%
3	177	F	3	60.0%	11.1%
4	235	F	4	71.9%	19.3%
5	309	S	-	----	----
6	386	S	-	----	----
7	428	F	5.67	86.2%	36.4%
8	428	S	-	----	----

For example, the rank values for specimen two indicate that we can be 90% confident that at least 4.7% to 47.1% of the population will fail on or before 119 cycles. To plot this information, mark the intersection of the plotted line with the percentage values corresponding to the two ranks selected. Pass vertical lines through these two rank positions on the Weibull line and find the points of intersection of these lines with a horizontal line passing through the plotted point for these specimens. These two intersection points define points of the 90% confidence interval. Continuing this procedure for the remaining failures and connecting the points obtained with smooth curves forms the 90% confidence band.

TABLE I-4. Table of Median Ranks [14]. Values for Sample Sizes Greater Than Ten Are Given in Reference [14].

Median Ranks (50% Ranks)

Sample Size = N

i	1	2	3	4	5	6	7	8	9	10
1	.5000	.2929	.2063	.1591	.1294	.1091	.0934	.0830	.0741	.0670
2		.7071	.5000	.3864	.3147	.2655	.2295	.2021	.1806	.1632
3			.7937	.6136	.5000	.4218	.3648	.3213	.2871	.2594
4				.8409	.6853	.5782	.5000	.4404	.3935	.3557
5					.8706	.7345	.6352	.5596	.5000	.4519
6						.8909	.7705	.6787	.6065	.5481
7							.9057	.7979	.7129	.6443
8								.9170	.8194	.7406
9									.9259	.8368
10										.9330

TABLE I-5. Tables of 5% and 95% Ranks [3]. Reference [3] Gives Values for Sample Sizes Larger Than Ten.

(a) Table of 5% Ranks

Sample Size = N

j	1	2	3	4	5	6	7	8	9	10
1	.0500	.0253	.0170	.0127	.0102	.0085	.0074	.0065	.0057	.0051
2		.2236	.1354	.0976	.0764	.0629	.0534	.0468	.0410	.0368
3			.3684	.2486	.1893	.1532	.1287	.1111	.0978	.0873
4				.4729	.3426	.2713	.2253	.1929	.1688	.1500
5					.5493	.4182	.3413	.2892	.2514	.2224
6						.6070	.4793	.4003	.3449	.3035
7							.6518	.5293	.4504	.3934
8								.6877	.5709	.4931
9									.7169	.6058
10										.7411

TABLE I-5 (continued).

(b) Table of 95% Ranks

Sample Size = N

j	1	2	3	4	5	6	7	8	9	10
1	.9500	.7764	.6316	.5271	.4507	.3930	.3482	.3123	.2831	.2589
2		.9747	.8646	.7514	.6574	.5818	.5207	.4707	.4291	.3942
3			.9830	.9024	.8107	.7287	.6587	.5997	.5496	.5069
4				.9873	.9236	.8468	.7747	.7108	.6551	.6076
5					.9898	.9371	.8713	.8071	.7486	.6965
6						.9915	.9466	.8889	.8312	.7776
7							.9926	.9532	.9032	.8500
8								.9935	.9590	.9127
9									.9943	.9632
10										.9949

TABLE I-6. Test Data and Information for Weibull Analysis of Hose No. 2

Specimen Number	Cycles	Survive/Fail	Rank	Median Rank	95% Rank	5% Rank
5	1,953	F	1	6.7%	25.9%	0.5%
8	2,675	S	-	----	----	----
3	8,440	F	2.11	17.4%	40.7%	4.2%
1	12,790	F	3.22	28.1%	52.95%	10.1%
6	15,098	S	-	----	----	----
2	16,773	F	4.52	40.6%	65.4%	18.8%
4	25,935	F	5.82	53.1%	76.3%	28.9%
8	27,888	S	-	----	----	----
9	27.888	S	-	----	----	----
10	27.888	S	-	----	----	----

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APPENDIX J

BASIS FOR THE SINGLE LOAD SEQUENTIAL TEST PLAN

Sequential testing is a technique by which we test sample parts one item at a time, and after testing each item, ask ourselves: "Can we be sure enough to accept or reject this batch or lot on the information so far collected?"

The basis theory of sequential testing was developed at Columbia University under the leadership of Abraham Wald during World War II. Dr. Benjamin Epstein and M. Sobel have since developed an approximate sequential procedure, based on Wald's work, which is satisfactory for most practical purposes. The Epstein-Sobel plan has the form:

$$\frac{\text{Unsatisfactory Life}}{-h_1 + rs} < T < \frac{\text{Satisfactory Life}}{h_o + rs} \qquad \text{(J-1)}$$

Where: T = test time

r = number of failures occurring up to time T

h_1, h_o, s = numerical constants derived from the parameters of the test plan

and where the test plan parameters are:

α = Producer's Risk - (Probability that parts which are actually *good* will be *rejected* by the sampling plan.)

β = Consumer's Risk - (Probability that parts which are actually *bad* will be *accepted* by the sampling plan.)

θ_o = Desired mean time between failures - (This is the characteristic life of the desired Weibull distribution.)

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θ_1 = Unacceptable mean time between failures - (This is the characteristic life of the unacceptable Weibull distribution.)

The sequential plans developed by Wald and Epstein-Sobel were based upon the exponential distribution. In order to extend the Epstein-Sobel plan to the Weibull distribution, a substitution must be made of θ^b for θ (where b = Weibull slope) in the numerical constants which are:

$$h_1 = \frac{\ln\left(\frac{1-\beta}{\alpha}\right)}{\frac{1}{\theta_1} - \frac{1}{\theta_o}} \qquad \text{(J-2)}$$

$$h_o = \frac{\ln\left(\frac{1-\alpha}{\beta}\right)}{\frac{1}{\theta_1} - \frac{1}{\theta_o}} \qquad \text{(J-3)}$$

$$s = \frac{\ln\left(\frac{\theta_o}{\theta_1}\right)}{\frac{1}{\theta_1} - \frac{1}{\theta_o}} \qquad \text{(J-4)}$$

The resulting test plan expressions are:

$$\overbrace{\frac{\theta_o^{\,b}}{\left(\frac{\theta_o}{\theta_1}\right)^b - 1} \times \left[r \times b \times \ln\left(\frac{\theta_o}{\theta_1}\right) - \ln\left(\frac{1-\beta}{\alpha}\right)\right]}^{\text{Unsatisfactory Life}} < T^b < \overbrace{\frac{\theta_o^{\,b}}{\left(\frac{\theta_o}{\theta_1}\right)^b - 1} \times \left[r \times b \times \ln\left(\frac{\theta_o}{\theta_1}\right) + \ln\left(\frac{1-\alpha}{\beta}\right)\right]}^{\text{Satisfactory Life}} \qquad \text{(J-5)}$$

For the sequential test plan, the following values and symbol were selected:

$$\alpha = \beta = 10\% = 0.10$$

$$\frac{\theta_o}{\theta_1} = \gamma$$

Substituting these into the above expressions and simplifying, we obtain the expressions for satisfactory and unsatisfactory life.

$$\overbrace{\frac{\theta_o^b}{(\gamma)^b - 1} \times \left[(r \times b \times \ln \gamma) - 2.1972\right]}^{\text{Unsatisfactory Life}} < T^b < \overbrace{\frac{\theta_o^b}{(\gamma)^b - 1} \times \left[(r \times b \times \ln \gamma) + 2.1972\right]}^{\text{Satisfactory Life}} \quad \text{(J-6)}$$

APPENDIX K

A BLUEPRINT FOR DEVELOPING RECOMMENDED STANDARD TEST PROCEDURES

FOREWORD

The document was released in this abbreviated form to provide discussion material for NFPA Project Group T2.6.1 R1. The thoughts presented in this paper are based on observations by the author and his colleagues made during the successful promulgation of recommended standards from inception through the International Standardization Organization. Some of these thoughts were documented in a letter by the author, dated 17 January 1977, which was sent to the ANSI/B93 Ad Hoc Committee on Pressure Rating.

INTRODUCTION

Recommended Standard Procedures are instrumental in reducing product costs, insuring product improvement, recording engineering technology, directing engineering research, and providing continued education for the industry. These extremely valuable documents require a commensurate amount of effort in the form of research, experimentation, creativity, and sales.

The staff of the Fluid Power Research Center has been involved in the development and promulgation of over 150 procedural documents with such organizations as the National Fluid Power Association, The Society of Automotive Engineers, The American Society of Lubrication

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Engineers, The International Association of Drilling Contractors, The American Society for Testing and Materials, The U.S. Army, The U.S. Navy, The National Aeronautics and Space Administration, The U.S. Air Force, and the International Organization for Standardization.

The purpose of this paper is to summarize the critical common factors associated with any standard procedure or test code development. It may not be necessary to follow every recommended step to obtain meaningful documents; however, the completion of each step increases the probability of the document receiving wide acceptance.

TEST CODE DEVELOPMENT BLUEPRINT

Figure 1 summarizes the procedure for developing a test code. The principle steps are listed below:

1. Formalize a title, scope, and purpose.
2. Select the best solution or solutions to achieve the desired goal. Sometimes, it is necessary to *"divide and conquer."* It may not be reasonable, practical, or pragmatic to prepare a *"get it all in one"* document.
3. Define acceptable levels of document reproducibility, discrimination, and accuracy.
4. Draft a procedure and a verification plan.
5. Obtain the assistance of *"outsiders,"* including recognized *"experts"* and interested knowledgeable friends, to review the

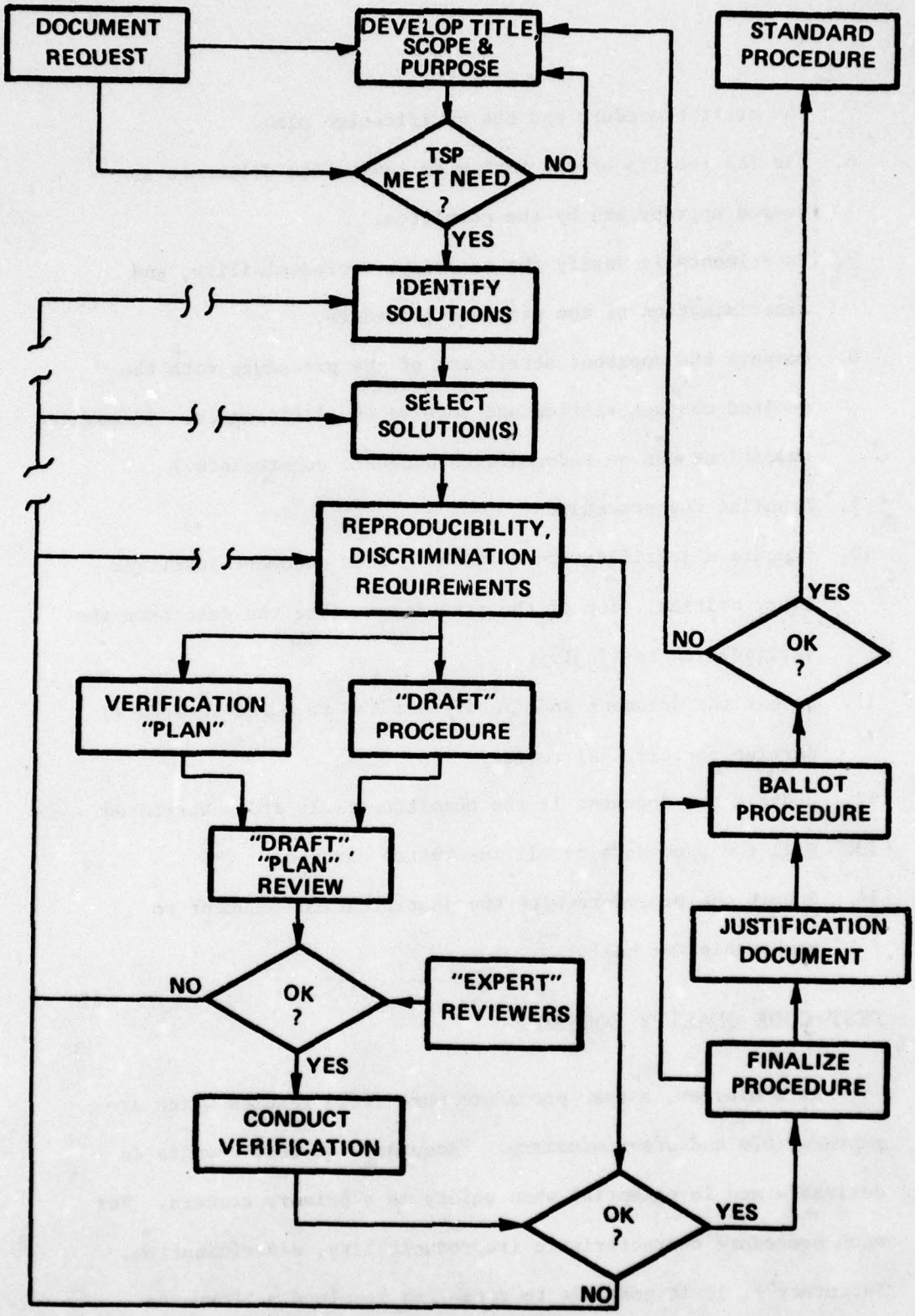


Fig. K-1. Abbreviated "Flow Chart" for Standard Test Procedure Development

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the draft procedure and the verification plan.

6. Use the results of the review to revise the documents as deemed appropriate by the committee.
7. Experimentally verify the accuracy, reproducibility, and discrimination of the proposed procedure.
8. Compare the apparent attributes of the procedure with the desired characteristics and resolve any differences. (Excessive variations can be reduced with adequate constraints.)
9. Finalize the procedure.
10. Prepare a justification document. This document justifies every critical step of the procedure. (Use the data from the verification testing.)
11. Submit the document and *"justification"* to a few interested parties for critical review.
12. Recycle the document if the committee feels it is warranted.
13. Sell the procedure to all interested parties.
14. Submit the procedure with the justification document to membership for ballot.

TEST CODE QUALITY CONTROL

As a minimum, a test procedure must yield results which are *reproducible* and *discriminatory*. "Accuracy" of test results is desirable and is essential when safety is a primary concern. For each procedure characteristic (reproducibility, discrimination, "accuracy"), it is possible to establish required performance

objectives. For this discussion, without assigning numerical values to these characteristics and within the realm of practical considerations, it should be sufficient to note the better these attributes, the better the test procedure. The only way that these attributes can be evaluated is by conducting tests with the procedure and analyzing the test results.

Those individuals who express genuine technical concerns about a test procedure are usually interested in assuring that the procedure yields acceptable levels of accuracy, discrimination, and reproducibility. *Adequate* procedural *constraints* will minimize the deviation of test results (and could eliminate some concerns). The validity of a test procedure and the significance of technical concerns can be properly examined by adequate experimental verification of a proposed test code.

CLOSURE

The approach for developing standard procedures, outlined in this appendix, has been shown to be effective for numerous test codes that are currently ISO procedures or are about to become ISO documents. The FPRC staff has had the frustrating experience of being associated with test codes that reached the ISO level without adequate experimental verification. If it is desired to produce meaningful test codes, they must be properly verified so that they are defendable as "state of the art" either at the ISO level or in the "court room."

The cardinal rule for standard procedure development and improvement is: "Seek the truth....incorporate it in the document and when in doubt....constrain the procedure."

SECTION II

NOISE ATTENUATION

PROJECT PERSONNEL

Dr. G. E. Maroney, Project Manager

Mr. D. L. O'Neal, Project Engineer

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PREFACE

This section presents a detailed account of the project activities in the area of hydraulic noise. Specific test procedures are recommended for determining the fluid-borne noise generation potential of hydraulic pumps, the airborne noise of hydraulic motors, and the acoustical performance characteristics of fluid-borne noise attenuators.

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CHAPTER I

INTRODUCTION

Noise in hydraulic systems originates as structureborne and fluidborne energy [1], [2], [3]. This energy is ultimately emitted to machine operators and bystanders as airborne noise. The control of noise can be achieved by modifying the source, isolation at the source, isolation along the path between the source and receiver, or isolating the receiver. Frequently, a combination of these control techniques is used. The most desirable technique is to use components which inherently produce a minimum amount of structureborne and fluidborne energy. The process of selecting "*quiet*" components necessitates having standard test procedures for assessing the "*noisiness*" of competitive products.

This report summarizes several activities which help insure the availability of adequate standard test procedures for assessing the "*noisiness*" of fluid power components. The activities for this project include the following major items:

- Assisting with the development of an ISO fluid power motor airborne noise test code.
- Assisting NFPA with the development and validation of a test code to assess the fluidborne noise generation potential of fluid power pumps
- Submission of the MERADCOM-OSU Fluidborne Noise Attenuator Test Procedure to industry for review.

These activities are discussed in Chapters II, III and IV respectively. The Appendicies contain supporting documentation.

The procedures which are the products of this project have provided the following:

- The basis of an "Airborne Motor Test Code" which is being promulgated at the ISO level.
- The basis of a "Fluidborne Pump Test Code" which is being promulgated at NFPA.
- The transmittal of a "Fluidborne Attenuator Test Code" to industry for review and promulgation.

All of these activities provide industry with opportunities to produce "*quieter*" fluid power systems.

CHAPTER II

PUMP FLUIDBORNE NOISE EVALUATION

Appendicies A and B contain two different test codes for evaluating the fluidborne noise generation potential of fluid power pumps. There are numerous possible ways of measuring the potential of a pump to produce fluidborne noise. A few of the possible techniques are discussed in [4].

The procedure in Appendix A, August 1977, was directed toward obtaining both the "blocked-pressure" and the "impedance" of a fluid power pump. When it became apparent that industry was more oriented toward a "single-number" test code ("blocked pressure" only), the procedure of October 1977 was introduced.

Neither of the methods discussed in this report have been completely verified. It is suspected that the "August" procedure will, as a minimum, need the addition of a section to insure that the system downstream of the load is an "effective" anechoic termination. That is, that the system downstream of the load does not have any significant influence on the measurements.

The "October" procedure comes close to providing the type of procedure that seems most palatable to industry. The procedure provides a single number rating. Like all single number ratings it can be very misleading. A natural and necessary companion document would be a test

code for evaluating the impedance of the pump. The results of recent tests by Morrison and McFatridge [5] again verify that the most realistic way to establish the load impedance (line size) is to make it relative to the pump impedance. Some measurement of the pump impedance is essential. The test code should be revised to reflect this fact.

CHAPTER III

MOTOR AIRBORNE NOISE EVALUATION

Appendix C contains a test code for determination of the airborne noise generated by hydraulic fluid power motors. A copy of this document was transmitted to ISO at the last meeting of ISO/TC131/SC8/WG1. This document served as the basis for a draft proposed, prepared by the referenced working group. It is anticipated that the draft document, because of its similarity to a pump airborne noise document, will be accepted by industry at the ISO level.

Once the document is available through ISO, the U.S. Army can use the test code as appropriate for controlling fluid power motor noise.

CHAPTER IV

FLUIDBORNE NOISE ATTENUATOR EVALUATION

References [1], [2], and [3] discuss the development of Fluidborne Noise Attenuator Test Codes. It should be emphasized that the attenuation characteristics of any "pressure ripple" reducer should be evaluated using a repeatable, discriminatory technique. The procedure contained in Appendix D provides repeatability and discrimination.

The procedure in the form of Appendix D has been transmitted to NFPA for consideration as a recommended procedure. The U.S. Army, as well as the rest of the industry using fluid power systems, should actively participate in the promulgation of an attenuator test code that provides discrimation and repeatability. Only after such a test code is available will fluid power system users be able to rationally select and design devices for reducing fluidborne noise.

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CHAPTER V

RECOMMENDATIONS

Previous reports [1], [2], and [3] discuss technical aspects of fluid power noise control, and the background for the test codes presented in this report. It is heartily recommended that the test codes presented in this report be pursued and promulgated until they become viable documents, readily available to the industry.

Fluid power system noise control will be attained when users and manufacturers adequately understand the causes of fluid power system noise. Standard test codes, their discussion, promulgation, and use provide a wealth of technology transfer which helps to insure a steady advancement of the "state of the art."

A reduction of fluid power noise means two important things to the U.S. Army:

1. Increased survivability---a six decibel reduction in the sound of a machine means that acoustic detection is twice as difficult.

2. Increased reliability---the presence of fluidborne noise in a fluid power system frequently becomes apparent because the "noise" causes early fatigue failures. A reduction of a system's fluidborne noise reduces the probability of failure. Elimination of the "noise" eliminates a potential cause of failure.

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Although many of the recommendations outlined in the previous reports [1] [2] [3] still stand, there are two critical areas that should be pursued of by MERADCOM:

1. Develop specific vibration isolation techniques that are directly applicable to fluid power systems. Demonstrate their usefulness. Put them into service.

2. Develop a design manual for Quiet Fluid Power Systems.

APPENDIX A

PROPOSED RECOMMENDED STANDARD
FOR MEASURING THE FLUIDBORNE NOISE CHARACTERISTICS
OF HYDRAULIC FLUID POWER PUMPS USING TWO HIGH IMPEDANCE LOADS
(TWO NUMBER TEST CODE)

REFERENCES

1. "*American National Standard Glossary of Terms of Fluid Power*," B93.2, American National Standards Institute.

2. "*American National Standard for Acoustical Terminology*," S1.1, American National Standards Institute.

3. "*SI Units and Recommendations for the Use of Their Multiples and of Certain Other Units*," ISO 1000, International Organization for Standardization, 1973.

4. Appendix to NFPA Recommended Standard Glossary of Terms of Fluid Power, T2.70.1.

5. "*Graphical Symbols for Hydraulic and Pneumatic Equipment and Accessories for Fluid Power Transmissions*," ISO/R 1219-1970.

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MEASURING THE FLUIDBORNE NOISE CHARACTERISTICS OF HYDRAULIC FLUID POWER PUMPS USING TWO HIGH IMPEDANCE LOADS

INTRODUCTION

In hydraulic fluid power systems, power is transmitted and controlled through a liquid under pressure within an enclosed circuit. Pumps are components which convert rotary mechanical power into fluid power. During the process of converting mechanical power into fluid power, airborne noise, fluidborne noise, and structureborne noise are generated by the pump. Fluidborne noise in hydraulic systems contributes to the overall system noise level and shortens the lives of system components. To accurately predict the noise generated by a particular positive displacement pump in a system, it is necessary to quantify the fluidborne noise characteristics of the pump. The fluidborne noise caused by a pump in a known system can be determined once the flow ripple (free volume) and the source impedance are known.

1. SCOPE

To include test procedure and methods of reporting hydraulic pump fluidborne noise generation characteristics. Specifically, this standard presents a uniform method for the determination of the free volume flow ripple and the source impedance of a pump for a given pump speed (or fluid flow), pump inlet pressure, pump outlet pressure, hydraulic fluid, and inlet fluid temperature. Hy-

draulic pumps evaluated using this standard may be of any type having all significant frequency components of fluidborne noise below 6000 Hz.

2. PURPOSE

To promote quieter and more reliable fluid power systems by establishing a uniform standard for measuring and reporting the fluidborne noise characteristics of hydraulic pumps.

3. TERMS AND DEFINATIONS

3.1 For definition of fluid power terms used in this document, see [1] and [4].

3.2 For definition of acoustical terms used in this document, see [2].

4. UNITS

The International System of Units (SI) is used in accordance with [3]. All measurements in dB are relative to 20 μPa.

5. SYMBOLS

5.1 Letter Symbols:

c	sonic velocity
d_o	pump outlet port diameter
j	$\sqrt{-1}$
L	test section length
$P_1(w)$	measured acoustic pressure with Load System 1 (see Appendix A-1)

$P_2(w)$	measured acoustic pressure with Load System 2 (see Appendix A-1)
$P_B(w)$	"blocked pressure"
$Q_f(w)$	"free volume" flow ripple
S_1	cross-sectional area, Test Section 1
S_2	cross-sectional area, Test Section 2
$Z_s(w)$	pump impedance

5.2 Graphic symbols used are in accordance with [5].

6. INSTRUMENTATION

6.1 Use standard measurement devices to measure fluid flow, static fluid pressure, pump speed, and fluid temperature in accordance with the recommendations for "industrial class" accuracy of testing specified in ISO ...

6.2 Select and maintain nonacoustic instrumentation such that measurements are accurate within the limits of Table A-1.

TABLE A-1. Measurement Accuracy.

Quantity	Accuracy Within (±) of Actual Valve
Pressure (static)	2%
Flow	2%
Motor Speed	2%
Temperature	$0.5^{o}C$

6.3 Use a narrow band analyzer to measure fluidborne noise levels. Acoustic instrumentation to be calibrated in accordance with ISO ...

6.4 Acoustic pressure transducers to be calibrated in accordance with OSU ...

7. CONSTRUCTION OF TEST SYSTEM

7.1 Mount pump in a manner that will minimize vibration transmission to the pressure transducer to be installed near the pump outlet.

7.2 Construct and connect the hydraulic system sketched in Fig. 1.

7.2.1 Load system construction and installation.

7.2.1.1 Construct load systems (numbers 1 and 2) in accordance with the instructions in Appendix A-1.

7.2.1.2 Install a "load valve" for initial system operation. (See Clause 7.3.)

7.2.2 Include in the circuit all oil filters, oil coolers, reservoirs, and restrictor valves as required to meet the pump hydraulic operating conditions specified in Table 2.

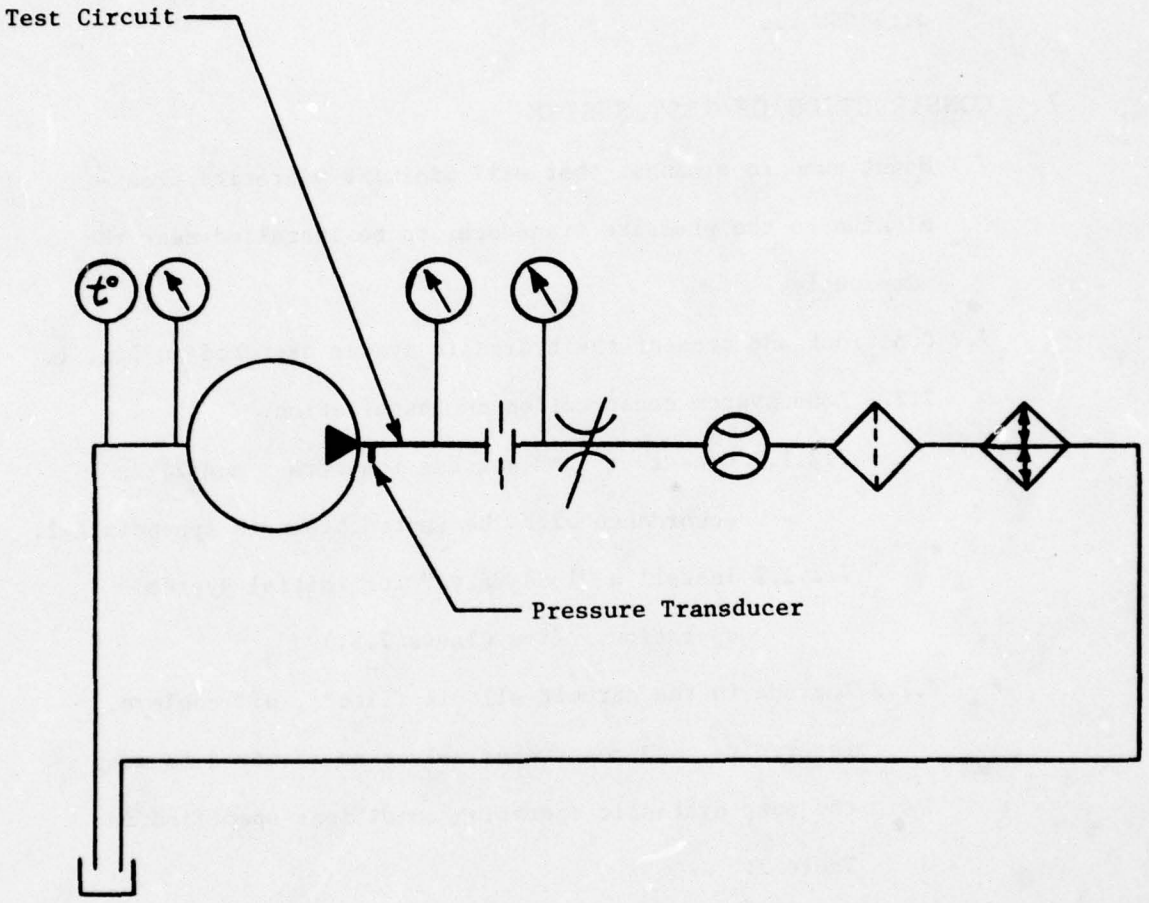


Fig. A-1. Example Test Circuit.

TABLE A-2. Allowable Variations in Test Circuits.

Test Parameter	Allowable Variation
Fluid Flow	± 2%
Fluid Pressure (static)	± 2%
Pump Speed	± 2%
Fluid Temperature	± 2°C

7.2.3 Use test fluid and filtration specified by the test request.

7.2.4 Use inlet line diameter in accordance with the manufacturers' recommended practice. Exercise extra care when assembling inlet lines to prevent air leaking into the circuit.

7.2.5 Mount the inlet pressure gauge at the same height as the inlet fitting or calibrate for any height difference.

7.3 "Run-in" the pump in accordance with manufacturer's recommendations before carrying out tests.

7.4 Prior to commencement of a series of tests, operate the pump for a sufficient time to purge air from the system and to stabilize all variables, including fluid condition, to within the limits specified in Table 2.

8. TEST PROCEDURE

8.1 Install Load System No. 1 and connect acoustic instrumentation.

8.2 Open load valve.

8.3 Adjust pump speed to obtain specified flow (or speed).

8.4 Close load valve to obtain specified pump outlet pressure. (Verify that the pressure ratio constraints of Appendix A-1 are met.)

8.5 Measure acoustic pressure as a function of frequency.

8.6 Label this acoustic pressure spectrum $P_1(w)$.

8.7 Repeat Steps 8.1 to 8.5 using Load System No. 2. Label the resulting spectrum $P_2(w)$.

8.8 Determine the sonic velocity in the fluid at the specified pump outlet pressure and specified operating temperature in accordance with OSU ...

8.9 Determine the fluid density of the fluid at the specified pump outlet pressure and specified operating temperature in accordance with OSU ...

9. CALCULATIONS

(At this point, the quantities ρ, C, $P_1(w)$, S_1, $P_2(w)$, and S_2 are known.)

9.1 Calculate the characteristic impedance of each test section:

$$Z_{01} = \rho C/S_1 \qquad Z_{02} = \rho C/S_2$$

9.2 Determine Q_f and Z_s at each frequency of interest. (These frequencies are the ten highest peak frequencies.)

9.2.1 Calculate the total load impedance on the pump for each test section:

$$Z_1 = -j\, Z_{01} \cot (wL/C) \qquad Z_2 = -j\, Z_{02} \cot (wL/C)$$

9.2.2 Calculate Z_s:

$$Z_s = (Z_1 Z_2 (P_2 - P_1)) \,/\, (P_1 Z_2 - P_2 Z_1)$$

NOTE: In these calculations, consider P_1, P_2, P_B, and Q_f as real values rather than complex numbers, since phase information for these quantities is not required.

9.2.3 Calculate P_B:

$$P_B = P_1 \left((Z_1 + Z_s)/Z_1\right)$$

9.2.4 Calculate Q_f:

$$Q_f = P_B / |Z_s|$$

10. DATA PRESENTATION

Compile and record the following information for all measurements made according to the requirements of this procedure:

10.1 General information

10.1.1 Name and address of pump manufacturer and, if applicable, user.

10.1.2 Reference number(s) for identification of pump.

10.1.3 Name and address of persons or organization responsible for the tests on the pump.

10.1.4 Date and place of tests.

10.2 Pump under test.

10.2.1 Type of pump (e.g., gear or piston) including ancillary equipment.

10.2.2 Type of displacement (e.g., fixed or variable).

10.2.3 Pump overall linear dimensions (with sketch if necessary).

10.2.4 Pump maximum displacement.

10.2.5 Type of displacement controller and setting.

10.2.6 Description of pump mounting conditions.

10.2.7 Nature and characteristics of the hydraulic circuit.

10.3 Instrumentation.

10.3.1 Details of equipment used to monitor pump operating conditions, including type, serial number, and manufacturer.

10.3.2 Details of equipment used for acoustic pressure measurements, including name, type, serial number and manufacturer.

10.3.3 Band width of frequency analyzer.

10.3.4 Overall frequency response of instrumentation system and date and method of calibration.

10.3.5 Method of calibration of pressure transducers and date and place of calibration.

10.4 Pump operating conditions. Include the following details for each test:

10.4.1 Type of fluid

10.4.2 Fluid viscosity (SUS or m^2/s)

10.4.3 Shaft speed (rad/s)

10.4.4 Inlet pressure (bar or kPa)

10.4.5 Outlet pressure (bar or kPa)

10.4.6 Pump delivery (flow) either measured or calculated (ℓ/s)

10.4.7 Temperature of fluid at pump inlet (^{o}C)

10.4.8 Fluid density

10.4.9 Sonic velocity in fluid at outlet pressure and operating temperature.

10.5 Report P_1, P_2, Z_1, Z_2, Z_s, P_B, and Q_f for the ten frequencies of interest, as shown in Table 3.

11. IDENTIFICATION

Use the following statement in catalogs and sales literature when electing to comply with this voluntary standard:

11.1 "Fluidborne noise characteristics of pump obtained and presented in accordance with NFPA Recommended Standard..."

Table A-3. Data Summary Sheet.

Testing Organization: _____ Date of Test: _____

Pump: Manufacturer _____ Type of Displacement _____
Identification _____ Maximum Displacement _____
Type _____ Displacement Tested _____
Overall Dimensions _____ Outlet Port Diameter _____

Fluid: Type _____ Density _____
Viscosity _____ Sonic Velocity _____

Test Conditions: Shaft Speed _____ Inlet Pressure _____
Volume Flow _____ Outlet Pressure _____
Inlet Temperature _____

Test Sections: L _____ D_1 _____ D_2 _____
S_1 _____ S_2 _____ Z_{01} _____ Z_{01} _____

No.	Freq.	P_1	P_2	Z_1	Z_2	Z_s	P_B	Q_f
1								
2								
3								
4								
5								
6								
7								
8								
9								
10								

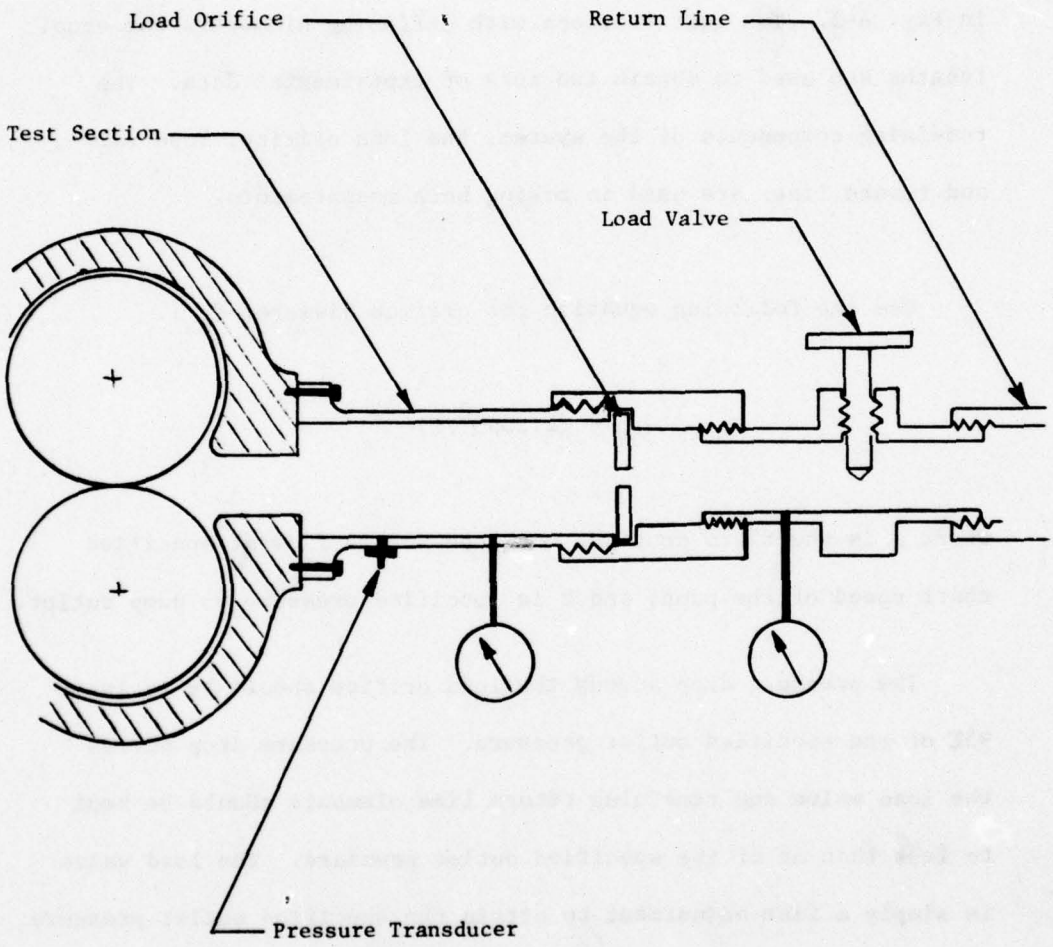


Fig. A-2. Example Load System.

APPENDIX A-1

CONSTRUCTION OF PUMP LOAD SYSTEM

The load system consists of four hydraulic components, a test section, a load orifice, a load valve, and a return line, as shown in Fig. A-1. Two test sections with differing diameters but equal lengths are used to obtain two sets of experimental data. The remaining components of the system, the load orifice, load valve, and return line, are used in making both measurements.

Use the following equation for orifice diameter d_{OR}:

$$d_{OR} = (2.25\rho Q^2/P)^{1/4}$$

where ρ is the fluid density, Q is the volume flow at specified shaft speed of the pump, and P is specified pressure at pump outlet.

The pressure drop across the load orifice should be at least 95% of the specified outlet pressure. The pressure drop across the load valve and remaining return line elements should be kept to less than 5% of the specified outlet pressure. The load valve is simply a fine adjustment to attain the specified outlet pressure. It may be necessary to change orifice sizes to achieve the 95% pressure drop across the load orifice.

The orifice should be sharp-edged with a thickness of 1/4 inch (6.4 mm) or less, but it should not be so thin as to cause significant fluctuation of the orifice due to pressure ripple.

The test sections should be constructed of thick-wall metal to prevent significant wall fluctuation due to pressure ripple. Make both test sections two inches long (measured from pump outlet face to upstream orifice edge). Use inside diameters of 1/2 d_o and 2 d_o, where d_o is the pump outlet port diameter.

APPENDIX B

PROPOSED RECOMMENDED STANDARD FOR MEASURING THE FLUIDBORNE NOISE CHARACTERISTICS OF HYDRAULIC FLUID POWER PUMPS (SINGLE NUMBER TEST CODE)

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REFERENCES

1. "*American National Standard Glossary of Terms for Fluid Power*," B93.2, American National Standards Institute.

2. "*American National Standard for Acoustical Terminology*," S1.1, American National Standards Institute.

3. "*SI Units and Recommendations for the Use of Their Multiples and of Certain Other Units*," ISO 1000, International Organization for Standardization, 1973.

4. Appendix to NFPA Recommended Standard Glossary fo Terms for Fluid Power, T2.70.1.

5. "*Graphical Symbols for Hydraulic and Pneumatic Equipment and Accessories for Fluid Power Transmissions*," ISO/R 1219-1970.

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MEASURING THE FLUIDBORNE NOISE CHARACTERISTICS

OF HYDRAULIC FLUID POWER PUMPS

INTRODUCTION

In hydraulic fluid power systems, power is transmitted and controlled through a liquid under pressure within an enclosed circuit. Pumps are components which convert rotary mechanical power into fluid power. During the process of converting mechanical power into fluid power, fluidborne noise is generated. Fluidborne noise in hydraulic systems contributes to both the structureborne and airborne system noise levels. To accurately predict the noise generated by a particular positive-displacement pump in a system, it is necessary to quantify the fluidborne noise characteristics of the pump. The fluidborne noise caused by a pump in a known system can be determined once any two of the following are known for the pump: free volume flow ripple, source impedance, or blocked acoustic pressure. Under most conditions, however, the blocked acoustic pressure alone may be used for quantitatively comparing the fluidborne noise generation potential of hydraulic pumps.

1. SCOPE

To include test procedures and methods for determining hydraulic power pump fluidborne noise at any specified operating conditions. Primarily, this standard provides for a single number fluidborne noise pressure rating. It also provides for standardized measurement of fluidborne noise pressure spectra up to 5000 Hz. Hydraulic pumps evaluated using this standard may be of any type having all significant frequency components of fluidborne noise below 5000 Hz.

2. PURPOSE

To promote quieter fluid power systems by establishing a uniform standard for measuring and reporting the fluidborne noise generation potential of hydraulic pumps.

3. TERMS AND DEFINITIONS

3.1 For definition of fluid power terms used in this document, see Ref. [1] and Ref. [4].

3.2 For definition of acoustical terms used in this document, see Ref. [2].

3.3 Blocked Acoustic Pressure (P_B) – the root-mean-square value of the oscillating component of pressure which a pump would develop if connected to an infinite load impedance.

3.4 Free Volume Flow Ripple (Q_f) – the root-mean-square value of the oscillating component of flow which a pump would develop if a zero impedance load was placed on the pump's outlet.

3.5 Source Impedance (Z_s) – the internal characteristic impedance associated with a pump.

3.6 Load Impedance (Z_L) – the impedance seen by the pump at its outlet port.

4. UNITS

4.1 The International System of Units (SI) is used in accordance with Ref. [3].

4.2 All pressure measurements in dB are relative to 20 μPa.

5. SYMBOLS

5.1 Letter Symbols:

c	sonic velocity in the fluid
d_o	pump outlet port diameter
j	$\sqrt{-1}$
P_B	blocked acoustic pressure
Q_f	free volume flow ripple
Z_v	load impedance
Z_o	acoustic impedance of the load system hose
Z_s	source impedance
Z_v	acoustic impedance of the load valve

5.2 Graphic symbols used are in accordance with Ref. [5].

6. INSTRUMENTATION

6.1 Use standard measurement devices to measure fluid flow, static fluid pressure, pump speed, and fluid temperature in accordance with the recommendations for "industrial class" accuracy of testing specified in ISO ...

6.2 Select and maintain non-acoustic instrumentation such that measurements are accurate within the limits of Table 1.

TABLE B-1. Measurement Accuracy

Quantity	Accuracy within (±) of Actual Value
Pressure (static)	2%
Flow	2%
Motor Speed	1%
Temperature	0.5°C

6.3 Use a narrow-band analyzer to measure fluidborne noise levels. Acoustic instrumentation to be calibrated in accordance with ISO ...

6.4 Acoustic pressure transducers to be calibrated in accordance with O.S.U...

7. GENERAL PROVISIONS

7.1 Mount pump in a manner that will minimize vibration transmission to the pressure transducer to be installed near the pump outlet.

7.2 Construct and connect the hydraulic load system to the pump, as shown in Fig. 1. Refer to Section 8 for details of construction and methods of qualifying the load system.

7.3 Include in the circuit all oil filters, oil coolers, reservoirs, and restrictor valves as required to meet the pump hydraulic operating conditions specified in Table 2.

TABLE B-2. Allowable Variations in Test Conditions.

Test Parameter	Allowable Variation
Fluid Pressure (static)	± 2%
Pump Speed	± 1%
Fluid Temperature	± 2°C

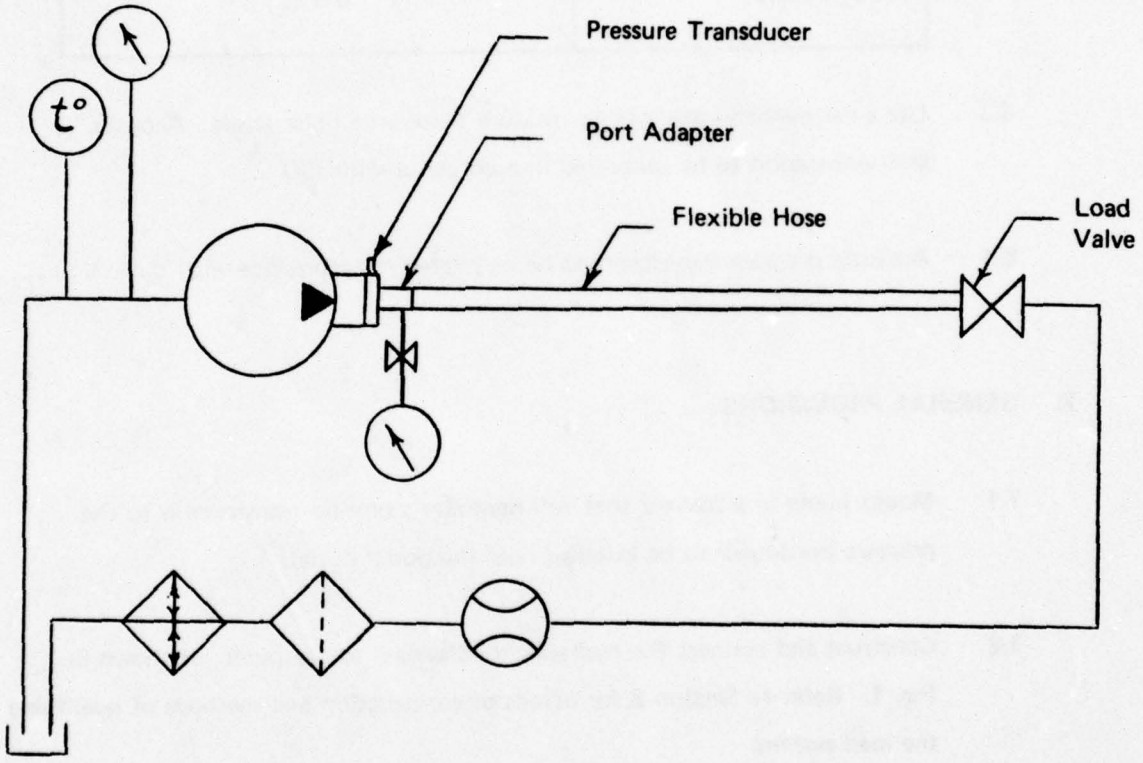


Fig. B-1. Example Test Circuit.

7.4 Use test fluid and filtration specified by the test request.

7.5 Use inlet line diameter in accordance with the manufacturers' recommended practice. Exercise extra care when assembling inlet lines to prevent air leaking into the circuit.

7.6 Mount the inlet pressure gauge at the same height as the inlet fitting or calibrate for any height difference.

7.7 "Run-in" the pump in accordance with manufacturer's recommendations before carrying out tests.

7.8 Prior to commencement of a series of tests, operate the pump for a sufficient time to purge air from the system and to stabilize all variables, including fluid condition, to within the limits specified in Table 2.

8. LOAD SYSTEM

8.1 Construct the load system as shown in Fig. 2.

8.1.1 Mount the pressure transducer as close as practical to the pump; mounting in the port adapter is recommended.

8.1.2 Mount the pressure transducer such that its sensing surface is flush with the inner diameter of the port adapter.

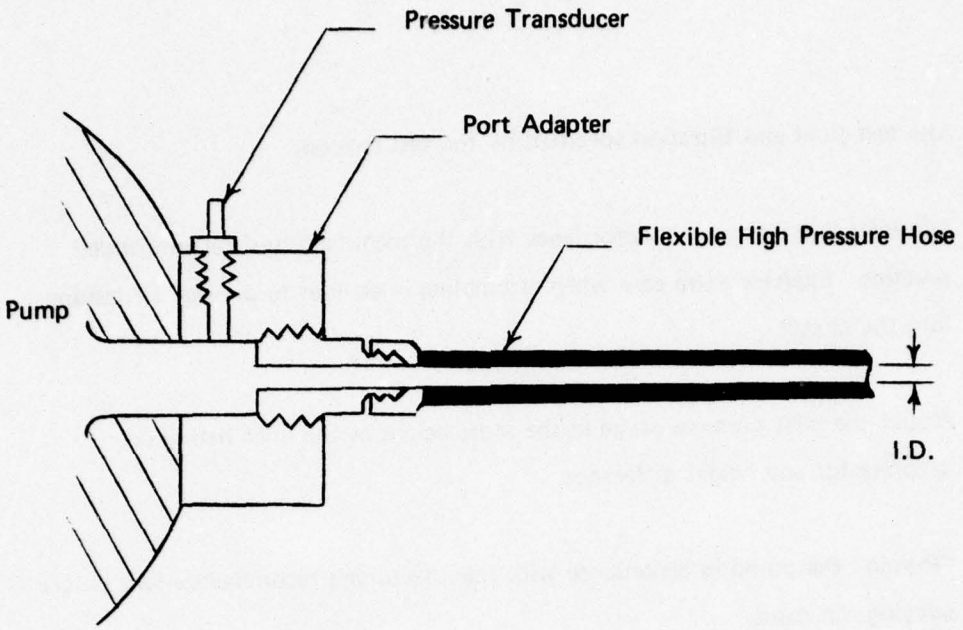


Fig. B-2. Example Load System.

8.1.3 The port adapter and the flexible hose should have the same inner diameter, with the transition from adapter to hose continuous. The diameter for the given flow, based on a fluid velocity of 9.1 m/sec (30 ft/sec):

$$ID = 0.153\sqrt{Q}$$

where ID is in cm, Q is in ℓ/min, or:

$$ID = 0.117\sqrt{Q}$$

where ID is in inches and Q is in gal/min.

8.1.4 The hose should be long enough to provide sufficient damping of resonant frequencies. A length of 2 m (6 ft) is suggested,

but this may be lengthened to achieve an anechoic line, as discussed in 8.2

8.1.5 Use a variable area load valve without free-moving mechanical parts.

8.1.6 Install a valve on the line to the outlet port pressure gauge to facilitate closing off the circuit when the gauge is not in use; this valve should be as close to the load system as possible.

8.1.7 It is important to establish that the test fluid is free from entrained air. Check all lines and connections to insure that no air is being drawn into the system. It is recommended that the test system reservoir be provided with an inspection opening so that the test fluid may be examined visually to insure that no entrained air is present.

8.2 The following procedure should be conducted to verify that the line is sufficiently "anechoic" to damp out resonances:

8.2.1 Construct the test system shown in Fig. 3; the inside diameter of the tubing on which the six transducers are mounted should be the same as the hose.

8.2.2 Operate the pump at the given operating conditions.

8.2.3 Record pressure transducer measurements as a function of frequency in dB.

8.2.4 Insure that the range of dB readings is less than 4 dB for the frequency range of interest.

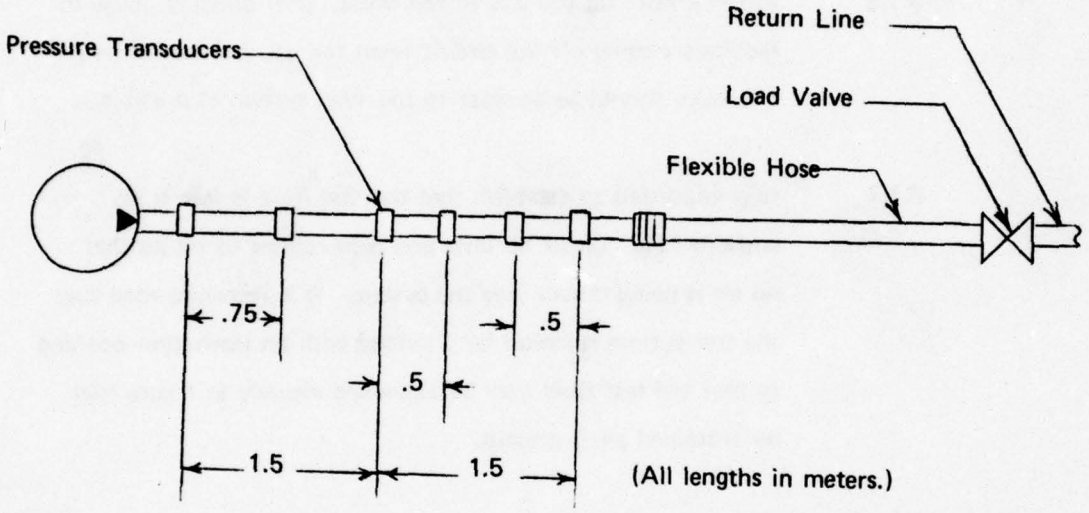


Fig. B-3. System for Verification of Anechoic Properties.

9. TEST PROCEDURE

9.1 Install load system and connect acoustic instrumentation.

9.2 Open load valve.

9.3 Adjust pump speed to obtain specified flow (or speed).

9.4 Close load valve to obtain specified pump outlet pressure.

9.5 Close valve in line to outlet pressure gauge.

9.6 Measure the "all-pass" acoustic pressure. This is the pump acoustic pressure rating.

9.7 Measure acoustic pressure as a function of frequency for the frequency range of interest. A component will be considered significant if it is within 10 dB of the maximum blocked pressure component observed. These pressures are the acoustic pressures at significant frequencies.

9.8 Determine the sonic velocity in the fluid at the specified pump outlet pressure and specified operating temperature in accordance with NFPA ...

9.9 Determine the density of the fluid at the specified pump outlet pressure and specified operating temperature in accordance with NFPA ...

10. DATA PRESENTATION

Compile and record the following information for all measurements made according to the requirements of this procedure:

10.1 General Information

10.1.1 Name and address of pump manufacturer and, if applicable, user.

10.1.2 Reference number(s) for identification of pump.

10.1.3 Name and address of persons or organization responsible for the tests on the pump.

10.1.4 Date and place of tests.

10.2 Pump Under Test

10.2.1 Type of pump (e.g., gear or piston) including ancillary equipment.

10.2.2 Type of displacement (e.g., fixed or variable).

10.2.3 Pump overall linear dimensions (with sketch if necessary).

10.2.4 Pump maximum displacement.

10.2.5 Type of displacement controller and setting.

10.2.6 Description of pump mounting conditions.

10.3 Instrumentation

10.3.1 Details of equipment used to monitor pump operating conditions, including type, serial number, and manufacturer.

10.3.2 Details of equipment used for acoustic pressure measurements, including name, type, serial number, and manufacturer.

10.3.3 Band width of frequency analyzer.

10.3.4 Overall frequency response of instrumentation system and date and method of calibration.

10.3.5 Method of calibration of pressure transducers and date and place of calibration.

10.4 Pump Operating Conditions – Include the following details for each test:

10.4.1 Type of fluid.

10.4.2 Fluid viscosity (SUS or m^2/s).

10.4.3 Shaft speed (rad/s).

10.4.4 Inlet pressure (bar or kPa).

10.4.5 Outlet pressure (bar or kPa).

10.4.6 Pump delivery (flow) either measured or calculated (ℓ/s).

10.4.7 Temperature of fluid at pump inlet (°C).

10.4.8 Fluid density.

10.4.9 Sonic velocity in fluid at outlet pressure and operating temperature.

10.5 Load System – Include the following:

10.5.1 Type and dimensions of outlet flange.

10.5.2 Type and dimensions of hose.

10.5.3 Type of load valve.

10.5.4 Nature and description of remaining elements in the hydraulic circuit.

10.5.5 Details of qualification procedures on the load system.

10.6 Data Summary – Report the test data and additional information in Table 3.

11. IDENTIFICATION

Use the following statement in catalogs and sales literature when electing to comply with this voluntary standard:

11.1 "Fluidborne noise characteristics of pump obtained and presented in accordance with NFPA Recommended Standard ..."

TABLE B-3. Data Summary Sheet.

Testing Organization: _____ Date of Test: _____

Pump: Manufacturer _____ Type of Displacement _____

Identification _____ Maximum Displacement _____

Type _____ Displacement Tested _____

Overall Dimensions _____ Outlet Port Diameter _____

Fluid: Type _____ Density _____

Viscosity _____ Sonic Velocity _____

Test Conditions:

Shaft Speed _____ Inlet Pressure _____

Volume Flow _____ Outlet Pressure _____

Inlet Temperature _____

Load System:

Hose Dimensions (ID X OD X L) _____

Distance from Pump to Transducer _____

Distance from Pump to Load Valve _____

Test Data:

Pump Acoustic Pressure Rating _____

ACOUSTIC PRESSURES AT SIGNIFICANT FREQUENCIES:

No.	1	2	3	4	5	6	7	8	
Frequency									
P_B									

APPENDIX C

HYDRAULIC FLUID POWER - MOTORS - TEST CODE FOR DETERMINATION OF AIRBORNE NOISE LEVELS

Hydraulic fluid power – Motors – Test code for determination of airborne noise levels

0 INTRODUCTION

In hydraulic fluid power systems, power is transmitted and controlled through a liquid under pressure within an enclosed circuit. Motors are components that convert fluid power to mechanical power. During the process of converting hydraulic fluid power into mechanical power, airborne noise, fluid-borne vibrations and structure-borne vibrations are radiated from the motor.

The airborne noise level of a fluid power motor is an important consideration in component selection. The noise measurement technique must, therefore, be such as to yield accurate appraisals of these airborne noise levels. The determination of noise levels is complicated by the interactions which occur during noise measurements. The fluid-borne and structure-borne vibrations from the motor can be transmitted to the circuit and ultimately give rise to background airborne noise levels which could affect the determination of the motor airborne noise levels.

The procedures described in this document are intended to measure only the airborne noise radiated directly from the motor under test.

1 SCOPE

This document establishes a test code describing procedures based on ISO 2204 for the determination of the sound power levels of a hydraulic motor under controlled conditions of installation and operation, suitable for providing a basis for comparing the noise levels of motors in terms of:

- A-weighted sound power level;
- octave-band sound power levels.

From these sound power levels, reference sound pressure levels may be calculated for reporting purposes (see clause 13).

For general purposes, the frequency range of interest includes the octave bands with center frequencies between 125 and 8 000 Hz.[1])

2 FIELD OF APPLICATION

The test code specified by this document is applicable to all types of hydraulic fluid power motors operating under steady-state conditions, irrespective of size, except for any limitations imposed by the size of the test environment (see clause 6).

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1) 1 Hz = 1 s^{-1}

3 REFERENCES

ISO 2204, Acoustics – Guide to the measurement of airborne acoustical noise and evaluation of its effects on man.

ISO 3740, Acoustics – Determination of sound power levels of noise sources – Guidelines for the use of basic standards and for the preparation of noise test codes.(1)

ISO 3742, Acoustics – Determination of sound power levels of noise sources – Precision methods for discrete-frequency and narrow-band sources in reverberation rooms.

ISO 3743, Acoustics – Determination of sound power levels of noise sources – Engineering methods for special reverberation test rooms.(1)

ISO 3744, Acoustics – Determination of sound power levels of noise sources – Engineering methods for free-field conditions over a reflecting plane.(1)

ISO 3745, Acoustics – Sound power levels of noise sources – Precision methods for anechoic and semi-anechoic rooms.(1)

ISO 4409, Hydraulic fluid power – Positive displacement pumps, motors and integral transmissions – Methods of test under steady-state conditions.(2)

ISO 5590, Fluid power – Vocabulary.(2)

ISO ..., Hydraulic fluid power – Pumps, motors and integral transmissions – Errors and classes of measurement.(2)

IEC Publication 50 (08), International Electrotechnical Vocabulary (2nd edition) – Group 08: Electro/acoustics.

4 DEFINITIONS

4.1 free sound field: A sound field in a homogeneous, isotropic medium free of boundaries. In practice, it is a field in which the effects of the boundaries are negligible over the frequency range of interest.

4.2 free-field over a reflecting plane: A field produced by a source in the presence of one reflecting plane on which the source is located.

4.3 reverberant sound field: That portion of the sound field in a test room over which the influence of sound received directly from the source is negligible.

4.4 anechoic room: A test room having boundaries which absorb essentially all of the incident sound energy over the frequency range of interest, thereby affording free-field conditions over the measurement surface.

4.5 special reverberant room: A room having reverberation time/frequency characteristics as specified in ISO 3743.

4.6 mean-square sound pressure: The sound pressure averaged in space and time on a mean-square basis. In

(1) At present at the stage of draft.
(2) In preparation.

practice, this is estimated by space and time averaging over a finite path length or over a number of fixed microphone positions.

4.7 mean sound pressure level: Ten times the logarithm to the base 10 of the ratio of the mean-square sound pressure to the square of the reference sound pressure. The weighting network or the width of the frequency band used should always be indicated; for example, A-weighted sound pressure level, octave band sound pressure level. The reference sound pressure is 20 μPa.[3] (Unit: decibel (dB)).

4.8 sound power level: Ten times the logarithm to the base 10 of the ratio of a given sound power to the reference sound power. The weighting network or the width of the frequency band used should always be indicated. The reference sound power is 1 pW.[4] (Unit: decibel (dB)).

4.9 volume of source under test: The volume of the noise source is the volume of the envelope of the whole pump under test.

4.10 For definitions of other terms used, see ISO 5598.

5 MEASUREMENT UNCERTAINTY

Measurements made in accordance with this document tend to result in standard deviations which are equal to or less than those specified in table 1.

TABLE C-1 — Standard deviation of sound power level determinations

Standard deviation (dB) for octave band centered on (Hz)					
Hz	125	250	500	1 000 – 4 000	8000
dB	5.0	3.0	2.0	2.0	3.0

The standard deviations of table 1 include the effects of allowable variations in the positioning of the measurement points and in the selection of any prescribed measurement surface, but exclude variations in the sound power output of the source from test to test. For a source which emits noise with a relatively "flat" spectrum in the 100 to 10 000 Hz range, the A-weighted sound power level is determined with a standard deviation of approximately 2 dB.

6 TEST ENVIRONMENT

6.1 Conduct tests in environments which provide "free-field over a reflecting plane" or "special reverberant" conditions which meet the environmental qualification requirements described in ISO 3743 and ISO 3744 (see table 2).

6.2 Conduct tests in "anechoic" or "reverberant" environments when more exacting environmental qualification procedures and measurement techniques, as specified in ISO 3742 and ISO 3745 (see table 2), are required.

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(3) $1\mu Pa = 10^{-6} N/m^2$

(4) $1pW = 10^{-12} W$

TABLE C-2 – Environmental qualification procedures

Method of Measurement	Test environment	Relevant International Standard	Qualification procedure of relevant standard
Engineering	Free-field over a reflecting plane	ISO 3744	Clause 4 and annex A
Engineering	Special reverberant	ISO 3743	Clause 4
Precision	Anechoic	ISO 3745	Clause 4 and annex A
Precision	Reverberant	ISO 3742	annex A

7 INSTRUMENTATION

7.1 Use instrumentation to measure oil flow, oil pressure, motor speed and oil temperature in accordance with the recommendations for "industrial class" accuracy of testing as specified in ISO ...

7.2 Use instrumentation for acoustical measurements in accordance with clause 4 of the relevant International Standard specified in table 2 for both performance and calibration.

8 INSTALLATION CONDITIONS OF MOTOR

8.1 Motor location

Locate the motor in any position consistent with the source installation and measurement surface (or microphone traverse) requirements specified in the relevant International Standard (see table 2) for the test environment being used.

8.2 Motor mounting

8.2.1 Construct the motor mounting so that it will minimize the sound radiated by the mounting as a result of motor vibrations.

8.2.2 Construct mounting bracket of high damping material or with sound damping and sound insulating material applied to the bracket as required.

8.2.3 Employ vibration isolation techniques, if needed, even if the motor is usually securely mounted.

8.2.4 Use flange mountings that are as small as practical to minimize interference with radiation of sound towards the shaft end of the motor.

8.3 Motor driven load

Locate the driven system outside the test space and drive the load through flexible couplings and a long shaft or isolate the load system in an enclosure.

8.4 Hydraulic circuit

8.4.1 Include in the circuit all oil filters, oil coolers, reservoirs and restrictor valves as required to meet the motor hydraulic operating conditions (see clause 9).

8.4.2 Use test fluid and filtration in accordance with the manufacturers' recommendation.

8.4.3 Install inlet and discharge line diameters in accordance with the manufacturers' recommended practice. Exercise extra care when assembling outlet lines to prevent air leaking into the circuit.

8.4.4 Mount the outlet pressure gauge at the same height as the outlet fitting or calibrate for any height difference.

8.4.5 Select the lengths of pipe between the motor and any control valves which minimize setting up standing waves in the lines which can increase the sound radiated from the motor.

8.4.6 Use a variable area load valve without free-moving mechanical parts.

NOTE – Such loading devices in the lines can generate and transmit sound through the fluid and piping which can emerge as airborne sound at the motor.

8.4.7 Position any control valves far from the motor, preferably outside the test room, to minimize the interaction.

8.4.8 Wrap all fluid lines and valves in the test space with acoustical materials, if required (see 11.1). Use material having a sound transmission loss of at least 10 dB at 125 Hz, and greater loss at higher frequencies.

9 OPERATING CONDITIONS

9.1 Determine the sound power levels of motors for any desired set of operating conditions (see 14.2.7).

9.2 Maintain these test conditions during the test within the limits specified in table 3.

TABLE C-3 – Allowable variations in test conditions

Test parameter	Allowable variation (±)
flow	2%
pressure	2%
speed	2%
temperature	2°C

9.3 Test the motor in the "as delivered" condition with any ancillary equipment operating normally during the test, so as to include their noise contributions to the airborne noise level of the pump.

10 LOCATION AND NUMBER OF SOUND MEASUREMENT POINTS

The location and number of measurement points are to be as required by the relevant International Standard stated in table 2 for the particular environment and method of measurement selected for the motor noise test.

11 TEST PROCEDURE

11.1 Background noise measurements

11.1.1 Measure the background noise of interest that is present during the motor noise test which does not emanate from the motor itself.

NOTE – Over the frequency range of interest, the band sound pressure levels of this background noise are to be at least 6 dB below the motor band sound pressure levels at each measurement point.

11.1.2 Correct for this background noise, if evidenced by measurement, by applying the corrections for this purpose given in the relevant International Standard in table 2.

11.1.3 When measuring band levels of background noise is not practical, the A-weighted background sound level of each measurement point is to be at least 6 dB below the motor A-weighted sound level.

11.1.4 Correct only A-weighted measurements for background noise. Easing the requirements for background noise levels can lead to an overestimate of the motor band sound pressure levels.

NOTE – The A-weighted background sound level at each measurement point may be checked by covering the pump with sound insulating materials capable of a transmission loss of at least 10 dB over the frequency range which is "determining" the A-weighted sound level of the motor.

11.1.5 If the background level is found to be too high, check for further noise control of the motor mounting, driven load, or hydraulic circuit as indicated.

11.1.6 Indicate the orientation of the microphone and the period of observation as specified in the relevant International Standard, see table 2.

11.2 Motor measurements

Prior to commencement of a series of tests, operate the motor for a sufficient time to purge air from the system and to stabilize all variables, including fluid condition, to within the limits specified in table 3.

Measure the following for each test:

a) Motor speed and flow rate;

b) Fluid temperature and pressure at motor inlet and fluid pressure at discharge fittings or at the test station provided by the motor manufacturer;

c) Band sound pressure levels at each measurement point over the frequency range of interest;

d) A-weighted sound level at each measurement point, if required by the relevant International Standard.

11.2.1 New or rebuilt motor

11.2.1.1 Repeat the first motor measurement test of the series at the end of a test series or after one hour of testing.

11.2.1.2 Invalidate the whole test series if the A-weighted sound level at any selected measurement point does not duplicate that of the first test within 2 dB(A).

12 CALCULATION OF MOTOR MEAN SOUND PRESSURE LEVELS AND SOUND POWER LEVELS

12.1 Correct the measured band sound pressure levels (and A-weighted sound levels where appropriate) at each measurement position for the measured background noise (background noise corrections).

12.2 Use these corrected levels to calculate the motor mean band sound levels and mean A-weighted sound level.

12.3 Calculate the motor sound power level from these mean sound pressure levels, taking into account any correction for unwanted environmental reflections (environmental correction factor).

12.4 Refer to the relevant International Standard as shown in table 2 to find information regarding corrections to be applied and the method of calculating the mean levels and pump sound power levels.

13 CALCULATION OF MEAN SOUND PRESSURE LEVEL AT A REFERENCE DISTANCE

Use the following equation to get the mean sound pressure level at a distance r (in metres) from the equivalent point source radiating into a free-field over a reflecting plane (hemispherical radiation) from the calculated motor sound power level:

$$\bar{L}_p = L_W - 10 \log 2\pi r^2/S_0$$

where:

$\bar{L}_p$ is the mean sound pressure level, A-weighted or in bands.

Reference: 20μPa;

L_W is the A-weighted or band power level of motor under test

Reference: 1 pW;

$2\pi r^2$ is the area of hemisphere of radius r (metres);

$S_0 = 1\ m^2$.

NOTE – For reporting purposes, choose a reference distance of r = 1m in which case the numerical value of $\bar{L}_p$ is obtained by subtracting 8 dB from the numerical value of the calculated sound power level L_W.

14 INFORMATION TO BE RECORDED

Compile and record the following information for all measurements made according to the requirements of this document:

14.1 General information

a) name and address of motor manufacturer and, if applicable, user;

b) reference number(s) for identification of motor

c) name and address of persons or organization responsible for the acoustic tests on the motor

d) date and place of acoustic tests;

e) statement that the sound power levels of the motor have been obtained in full conformance with this document and the relevant International Standard for the determination of sound power levels of noise sources as selected from table 2. (See also clause 16.)

14.2 Motor under test

14.2.1 Description of motor

a) type of motor including ancillary equipment (for example gear or piston);

b) type of displacement (for example fixed or variable);

c) motor overall linear dimensions (with sketch if necessary);

d) motor maximum displacement

e) type of displacement controller and setting.

14.2.2 Acoustic environment for tests

a) the test room internal dimensions and the type of acoustic field for the measurements (for example anechoic, free-field over a reflecting plane, reverberant or special reverberant);

b) the test room acoustic treatment;

c) the test room reverberation times (when applicable) and date of measurement;

d) ambient air temperature (in degrees Celsius), relative humidity (percentage) and barometric pressure (in millibars);

e) results of acoustical qualification of test environment as required by the relevant International Standard in table 2.

14.2.3 Reference sound source (when applicable)

a) manufacturer, type and serial number;

b) sound power level calibration data including name of calibrating laboratory and date of calibrations.

14.2.4 Mounting and installation conditions of motor

a) description of motor mounting conditions

b) nature and characteristics of the hydraulic circuit and details of any acoustic insulation treatment;

c) nature and description of other machines being used which could have an influence on the measured sound pressure levels of the motor

14.2.5 Location of motor in test environment

Include a sketch showing the location of the motor in relation to walls, floor and ceiling of test room. Also, show the location of other reflecting or absorbing screens and noise sources which can influence measurements.

14.2.6 Instrumentation

a) details of equipment used to monitor motor operating conditions (see 14.2.7), including type, serial number and manufacturer;

b) details of equipment used for acoustic measurements including name, type, serial number and manufacturer;

c) band width of frequency analyzer;

d) overall frequency response of instrumentation system and date and method of calibration;

e) method of calibration of microphones and date and place of calibration

14.2.7 Motor operating conditions

Include the following details for each test:

a) type of fluid;

b) fluid viscosity (SUS or m^2/s);

c) shaft speed (rad/s);

d) inlet pressure (bar or kPa);

e) outlet pressure (bar or kPa);

f) delivered flow (ℓ/s)

g) temperature of fluid at motor inlet (°C)

14.2.8 Acoustical data

Include all data as required by the relevant International Standard listed in table 2.

15 TEST REPORT

Provide the following information in the test report:

a) The A-weighted sound power level and octave band sound power levels for each frequency band of interest for each set of operating conditions.

b) A statement that the sound power levels have been obtained in full conformance with the procedures of this document and specific paragraphs of the relevant International Standard for the determination of sound power levels of noise sources as selected from table 2.

16 IDENTIFICATION STATEMENT (Reference to this document)

Use the following statement in test reports, catalogues, and sales literature when electing to comply with this document:

"Test code for the determination of airborne noise levels conforms to ISO .... , Hydraulic fluid power – Motors – Test code for the determination of airborne noise levels."

APPENDIX D

METHOD FOR EVALUATING THE PERFORMANCE OF A FLUIDBORNE NOISE ATTENUATOR

Note: This is a preliminary draft which does not include all references and table of contents.

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METHOD FOR EVALUATING THE PERFORMANCE OF A FLUIDBORNE NOISE ATTENUATOR

INTRODUCTION

In fluid power systems, power is transmitted and controlled through a fluid under pressure within an enclosed circuit. Pumps convert mechanical power into fluid power. Sound is created in the fluid during the power conversion process. In fluid power systems, fluidborne noise can be controlled with the use of fluidborne noise attenuators. The attenuator can reflect sound energy back toward the source, and absorb sound energy within it.

The need for a technique to evaluate the performance of fluidborne noise attenuators has long been recognized. One of the primary considerations in the selection of a fluidborne noise attenuator is its effectiveness in reducing the fluidborne noise in a system.

1. SCOPE

To include rating basis, test methods and method of reporting transmission loss, pressure drop, and impedance of fluidborne noise attenuators.

2. PURPOSE

To establish a uniform standard for measuring, reporting and accurately comparing the transmission loss, pressure drop, and impedance of fluidborne noise attenuators.

3. TERMS AND DEFINITIONS

3.1 For definition of Fluid Power terms used in this document, see Ref. 1.

3.2 For definition of Acoustical Terms used in this document, see Ref. 6.

3.3 Anechoic Termination. A termination in a hydraulic system which essentially reflects none of the incident sound energy over the frequency range of interest.

4. UNIT OF MEASUREMENT

The International System of Units (SI) is used in accordance with Ref. 5 dB relative to 20 μPa.

5. GRAPHIC SYMBOLS

Graphic symbols used are in accordance with Refs. 2,3 and 4. Where Refs. 3 and 4 are not in agreement with Ref. 2, Ref. 2 governs.

6. TEST EQUIPMENT

6.1 Use standard measurement devices to measure oil flow, oil pressure, pump speed and oil temperature.

6.2 Use one-third octave band or narrow band analyzer to measure fluidborne noise levels.

6.3 Use oscilloscope to measure sonic velocity.

6.4 Use oil as the test fluid.

7. TEST CONDITIONS ACCURACY

7.1 Maintain these test conditions during the test within the limits specified in Table D-1.

TABLE D-1. Test Conditions Accuracy

TEST CONDITION	CUSTOMARY U. S. UNIT	S. I. UNIT	MAINTAIN WITHIN (±) OF ACTUAL
FLOW	USGPM	ℓ/MIN	± 2%
PRESSURE	PSI	BAR	± 2%
SPEED	RPM	REV MIN	± 2%
TEMPERATURE	° F	° C	38 ± 2°C
SONIC VELOCITY	FT/SEC	M/SEC	± 5%

8. TEST PROCEDURE

8.1 Install Anechoic Termination

8.2 Construct Hydraulic Circuit

8.2.1 Install in the circuit oil filters, oil coolers, reservoirs, and control valve as needed downstream of the anechoic termination.

8.2.2 Use test fluid and filtration in accordance with the manufacturers' recommendation.

8.2.3 Install inlet and discharge line diameters in accordance with the manufacturers' recommended practice. Exercise extra care when assembling inlet lines to prevent air leaking into circuit.

8.3 Install Pressure Transducers

8.3.1 Install two sets of pressure transducers as shown in Fig. D-1.

8.3.2 Insure that transducer spacing is the same as in Fig. D-2 with spacing tolerance within ± 2%.

8.3.3 Place a section of test system, equivalent in length to the attenuator, between the two sets of pressure transducers.

8.4 Verify Anechoic Termination

8.4.1 Set system pressure to maximum test pressure for FBN attenuator.

8.4.2 Set speed of pump so that the fundamental pumping frequency is at 100 Hz.

8.4.3 Record pressure transducer measurements as a function of frequency in dB, for first three harmonics of fundamental pumping frequency.

8.4.4 Measure a "tare" pressure drop vs. flow rate.

8.4.5 Measure sonic velocity per OSU-FPRC procedure.

8.4.6 Repeat 8.4.2, 8.4.3, 8.4.4, and 8.4.5 for fundamental pumping frequencies of 125, 160, 267, and 315 Hz.

8.4.7 Insure that the range of dB readings ≥ 4 dB for all test frequencies.

8.4.8 Record all data in data table similar to Table D-2.

8.5 Measure Attenuator Performance

8.5.1 Remove the section of test system equivalent in length to the attenuator between the sets of pressure transducers.

8.5.2 Install FBN attenuator.

8.5.3 Repeat step 8.4.1.

8.5.4 Repeat step 8.4.2.

8.5.5 Repeat measurement in 8.4.3.

8.5.6 Measure flow and pressure drop across attenuator.

8.5.7 Relieve system pressure to a minimum pressure.

8.5.8 Increase system pressure to that in 8.5.3.

8.5.9 Repeat measurements in 8.5.5 and 8.5.6.

8.5.10 Repeat steps 8.5.7 and 8.5.8.

8.5.11 Repeat measurements in 8.5.9.

8.5.12 Repeat steps 8.5.4 up to and including 8.5.11 for fundamental pumping frequencies of 125, 160, 267, and 315 Hz as indicated in Table D-2.

8.6 Invert attenuator and repeat steps 8.5.3 through 8.5.12.

9. CALCULATIONS

9.1 Obtain pressure drop versus flow due to the attenuator by correcting 8.5.6 with 8.4.4.

8.2 Calculate the transmission loss and impedance of the attenuator from data taken in 8.5.

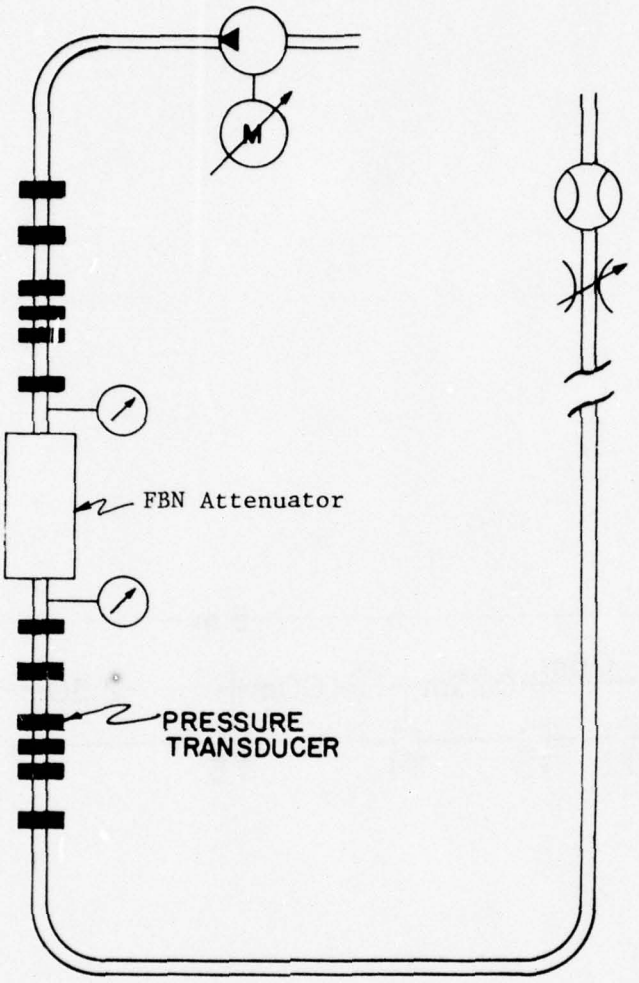
10. DATA PRESENTATION

10.1 Report pressure drop versus flow rate in a form similar to that in Fig. D-3.

10.2 Report attenuator transmission loss as a function of frequency as shown in Fig. D-2.

10.3 Report attenuator input reflection factor as a function of frequency as shown in Fig. D-3.

10.4 Report attenuator output reflection factor as a function of frequency as shown in Fig. D-3.



Step 1: Verify anechoic termination and measure "tare" P.
Step 2: Measure pressure drop across attenuator.
Step 3: Determine energy incident to attenuator $(p_i)^2$.
Step 4: Determine energy transmitted through attenuator $(p_i)^2$.
Step 5: Measure sonic velocity.
Step 6: T.L. = $20 \log_{10} p_i/p_t$.
Step 7: Calculate pressure drop

Fig. D-1. Summary of Test Procedure Showing Schematic of Measurement and Loading Circuit.

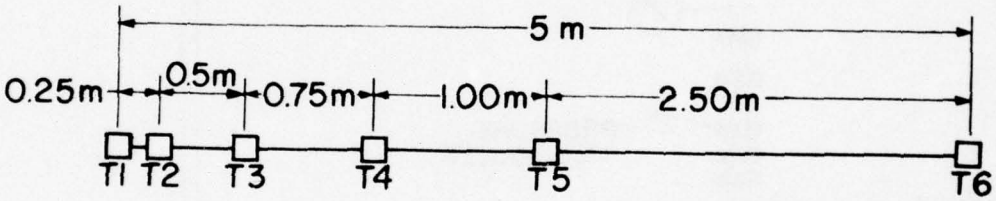


Fig. D-2. Transducer Spacing for FBN Attenuator Performance Evaluation

TABLE D-2. Data Table for FBN Attenuator Test.

Date:_____ Attenuator I.D._____ Pump I.D._____

Pump Speed:_____ System Pressure:_____ Flow Rate:_____

Fluid:_____ Fluid Viscosity: _____ Fundamental Pumping Frequency:_____

Tare Pressure Drop:_____ Pressure Diff._____

Harmonic	Transducer Location	Measurement	Pressure Level at Transducer (dB)						Sonic Velocity
			P_1	P_2	P_3	P_4	P_5	P_6	
f_1	Up	(1) (2) (3)							
		μ							
	Down	(1) (2) (3)							
		μ							
f_2	Up	(1) (2) (3)							
		μ							
	Down	(1) (2) (3)							
		μ							
f_3	Up	(1) (2) (3)							
		μ							
	Down	(1) (2) (3)							
		μ							

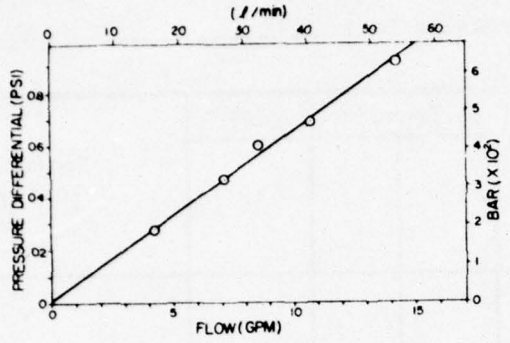


Fig. D-3. Pressure Drop Versus Flow Rate

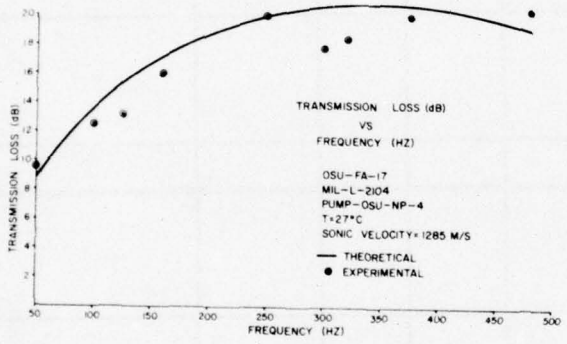


Fig. D-4. Transmission Loss Versus Frequency

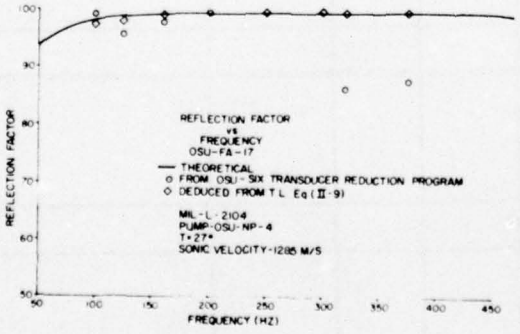


Fig. D-5. Reflection Factor as a Function of Frequency

11. SUMMARY OF DESIGNATED INFORMATION

The following designated information is needed when applying this recommended standard to a particular application or use.

11.1 Transmission loss spectrum

11.2 Pressure drop versus flow

11.3 Input reflection factor spectrum

11.4 Output reflection factor spectrum

APPENDIX E

REFERENCES

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REFERENCES

1. MERADCOM/OSU Hydraulic System Reliability Program, Annual Report, Contract No. DAAK02-72-C-0172 (1974).

2. MERADCOM/OSU Hydraulic System Reliability Program, Annual Report, Contract No. DAAK02-75-C-0137 (1976).

3. MERADCOM/OSU Hydraulic System Reliability Program, Annual Report, Contract No. DAAK02-75-C-0137 (1977).

4. O'Neal, D. L. and G. E. Maroney, *"Measuring Pump Fluidborne Noise Generation Potential,"* The BFPR Journal, 1978, 11:235-241.

5. Morrison, J. and D. McFatridge, *"Measurement of Fluidborne Noise,"* Research Report for Experimental Fluid Dynamics MAE 4273, Fluid Power Research Center, Oklahoma State University, May 1978.

APPENDIX F

ASSUMPTIONS FOR OCTOBER 1977
PUMP FLUIDBORNE NOISE TEST CODE

ASSUMPTIONS FOR OCTOBER 1977
PUMP FLUIDBORNE NOISE TEST CODE

(See Appendix B)

1. PUMP MODEL

It is assumed that the hydraulic pump and its FBN characteristics can be adequately represented by an electrical analog of an ideal source coupled with an ideal impedance.

2. P_b ALONE MAY BE USED FOR COMPARISON

At the August 24, 1977 NFPA meeting the simplicity of a single quantity was agreed upon as the most attractive course of action, perhaps adding information on determining Q_f or(Z_s) in an appendix. (Since then it has been suggested that a more complete set of characteristics should be obtained using the proposed standard.) The use of P_b alone as a means of comparison was considered adequate due to the similarity of FBN characteristics in most hydraulic systems.

3. MEASUREMENT ACCURACIES

The values shown in Table B-1 were obtained from air-borne noise standards currently used. Changes may be necessary.

4. ALLOWABLE VARIATIONS IN TEST CONDITIONS

The values shown in Table B-2 were obtained from air-borne noise standards currently in use. Changes may be necessary.

5. PORT ADAPTER SIZING

The inside diameter of the port adapter could also be determined by making it equal to the outlet port diameter upstream of the flange.

This was included to insure against arbitrarily picking an adapter size, not matching the pump, which could change FBN characteristics.

6. LOCATION OF TRANSDUCER

The transducer and small diameter hose should be as close as possible to the pump (length) to minimize the volume of the port adapter external to the pump.

7. HOSE TO ADAPTER TRANSITION

Obviously if the transition was a change in size, the acoustic properties would be more complex.

8. INNER HOSE DIAMETER

The problem here is deciding what should limit the inner diameter of the hose. The purpose of the load system is to provide an impedance as seen by the pump at its outlet that is much greater than the pumps internal impedance. Obviously the smaller the hose inner diameter is, the larger the acoustic impedance will be. Approximations for the source impedance and the characteristic impedance of a circular passage gave diameters for the hose too small to be practical. Experimental data showed that much larger hose works. More experimentation should determine the size of hose necessary and the parameters that determine. The proposed standard suggests using a size of hose to give a specified fluid velocity (30 ft/sec) which, at the time, appeared as good as anything else. However, experimentation with sizes should be done to determine the proper hose sizing procedure. (Later experimentation suggests that the line impedance should be about 10 times the pump impedance.)

9. NEED FOR DAMPING

A previously suggested method for measuring fluidborne noise consists of an orifice at the pump's outlet to simulate a blocked situation; the small diameter hose used in the procedure here is an improvement in that the hose acts as an anechoic termination to attenuate resonances.

10. PROPERTIES OF HOSE

The major consideration in choosing the type of hose is its damping ability. The method outlined for qualifying the hose was one used at the FPRC for qualifying "anechoic" terminations. The range of readings obtained is a measure of the magnitude of standing waves in the line, which in turn is a measure of the lines damping ability. Another consideration in choosing the type of hose is its flexibility.

11. RANGE OF READINGS

The valve 4 dB for the maximum allowable range was choosen because this valve was used for anechoic termination qualifications. Further work should be done to insure that this valve is adequate.

12. TRANSDUCER SPACING

The spacing shown is one used previously by FPRC.

13. ALL-PASS MEASUREMENT

This is the simplest single-number rating possible. A better quantity might be obtained by adding the major components together with some weighting scheme on the basis of frequency. The weighting would depend on the annoyance caused by FBN in each frequency range.

14. SPECTRAL COMPONENTS

All spectral components lower than the maximum minus 10 dB would be negligible unless the natural frequency of some part of the system falls nearby. In this case, all major components in the vicinity of a known natural frequency should be reported.

SECTION III

VALVE CONTAMINANT SENSITIVITY

PROJECT PERSONNEL

Dr. E. C. Fitch, Program Director

Dr. G. E. Maroney, Program Manager

Dr. S.K.R. Iyengar, Consultant

R. Sharp, Project Engineer

Mr. Bruce Larkin, Project Associate

PREFACE

This section presents a detailed account of the project activities in the valve contaminant sensitivity appraisal area. A specific test procedure to relief valves is developed and verified by experimental tests. Response to a questionnaire sent to the fluid power industry are also summarized and reviewed.

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CHAPTER I

INTRODUCTION

The reliability of a complex piece of machinery such as a mobile hydraulic system depends on the satisfactory performance of critical individual components. Almost all hydraulic systems rely on a relief valve to limit the circuit pressure and many of them contain additional relief valves to control pressures in local areas eg. cylinders, motors, etc. Experiments conducted at the Fluid Power Research Center, Oklahoma State University, have indicated that the static and dynamic performance of relief valves can change drastically after exposure to particulate contaminant, and that different designs of valves degrade to different extents [1] [2] [3]. Recognizing the importance of contaminant sensitivity appraisal, the Fluid Power Research Center developed apart of the MERADCOM-OSU Hydraulic Component Specification Program 1970-1972, a test procedure for relief valves [4] [5]. Though this test procedure could discriminate between valves with slow and rapid degradation rates [6], subsequent tests revealed that it did not give a complete picture of the effects of particulate matter on the valve characteristics. Consequently a revision of the test procedure was undertaken as part of the MERADCOM-OSU Hydraulic System Reliability Program for 1977-1978.

A comprehensive literature survey was conducted to ascertain not only the status of research in the area of contaminant sensitivity appraisal, but also test procedures used by users and manufacturers of relief valves. A draft procedure based on the experimental data collected at the Fluid Power Research Center and elsewhere was developed and

submitted to the industry for reviews. Simultaneously, an experimental validation program was initiated. Based on the inputs received from industry and experimental efforts, the final draft of the test procedure was developed.

CHAPTER II

RELIEF VALVE CONTAMINANT SENSITIVITY APPRAISAL

A relief valve is installed in a fluid power circuit to limit the pressure to a prescribed valve. An ideal relief valve would open at the set pressure, maintain that pressure irrespective of the flow rate through it, and respond instantaneously to changes in pressure [2]. It is obvious that no physical valve can meet all the above requirements. Almost all valves start passive flow at a pressure lower than the set pressure and exhibit a slope in the pressure-flow characteristics. It is very difficult to establish the pressure at which the valve just begins to pass flow, and this valve may show wide variation from test to test. Hence it is convenient to define the cracking pressure as that differential pressure at which the valve passes one-twentieth of the rated flow. Most valves also exhibit a positive gradient in the pressure flow characteristic. A large gradient implies that the systems pressure can change significantly as the flow rate changes. Since different systems require different degrees of control, the gradient or slope is vital information to the circuit designer. In summary it can be said that the cracking pressure and the slope constitute valve parameters of concern to valve manufacturers and end-item companies, as well as equipment users.

The objective of a contaminant sensitivity appraisal test is to measure and report the changes in the two parameters mentioned above. Unlike certain other hydraulic components, eg. pump and motors, it is not realistic to establish a single performance parameter since both the cracking pressure and the slope are significant as well as being independent. It

is conceivable that in certain applications, a valve showing large changes in cracking pressure is acceptable, while in others it is not. A similar situation may exist with respect to the slope.

A laboratory test for contaminant sensitivity appraisal has to be conducted under controlled conditions in order that comparisons and ranking can be made and accelerated, so as to be completed in reasonably short time. The generally accepted method of acceleration is that of using high concentrations of particulate contaminant, and the use of AC Fine Test Dust (ACFTD) ensures that the mode of degradation is similar to that naturally experienced by fluid power components on mobile equipment. Moreover, ACFTD has been used very successfully in testing components such as pumps, motors and cylinders, and from a logistics standpoint, it is advantageous to pursue the testing of valves above the same lines. It should be noted that the general procedure of testing, discussed below, could be followed for any kind of contaminant.

Even though the primary objective of contaminant sensitivity testing is to establish contaminant levels acceptable for satisfactory performance, an ultimate goal which must be kept in mind is to develop appropriate mathematical models which can be used to predict the degradation of a specific valve when exposed to a known environment. Hence, it is necessary to expose the test valve to contaminants of different particle sizes. Once again, in order to maintain uniformity with test procedures for other hydraulic components, "single-cut" ACFTD in size ranges 0-5μm through 0-80μm are used in the appraisal

The basic scheme of the test consists of measuring the cracking pressure and slope after subjecting the test valve sequentially to known contaminant levels of increasing size ranges. The remainder of the chapter explains the ramifications of the procedure needed to ensure repeatability and reproducibility. Reference will be made to paragraphs in the test procedure, which is included in Appendix A of this report.

Fig. II-1, a circuit schematic of the test system, consists essentially of a closed loop incorporating the reservoir, pump, test valve, and conditioning equipment. Flow meters, pressure gages/transducers, and thermometers. It is desirable to protect the flowmeter from contaminant exposure by suitable plumbing. The control filter should be capable of maintaining the system in the "clean" condition after repeated injections of contaminant.

Validation of the test facility, outlined in section 7.7, is a very important prerequisite to testing. By carefully designing the system, so as not to have contaminant traps or ingression and generation, and selecting suitable components, it can be ensured that contaminant levels can be maintained at prescribed levels. Hoses should be avoided, as also valves other than ball valves. Pressurized reservoirs are generally satisfactory, a cylindrical design with a conical bottom being one of the best designs. Stainless steel construction and tubing leads to a system with little built-in contaminants. The pump should be carefully selected so as to not only be contaminant insensitive but also not change the particle distribution. A variable speed drive is usually a convenient method of obtaining the desired flow rate. Fluid samples should be taken in such a manner that the resulting particle counts are representative of the test system. It is desirable to check the actual gravimetric

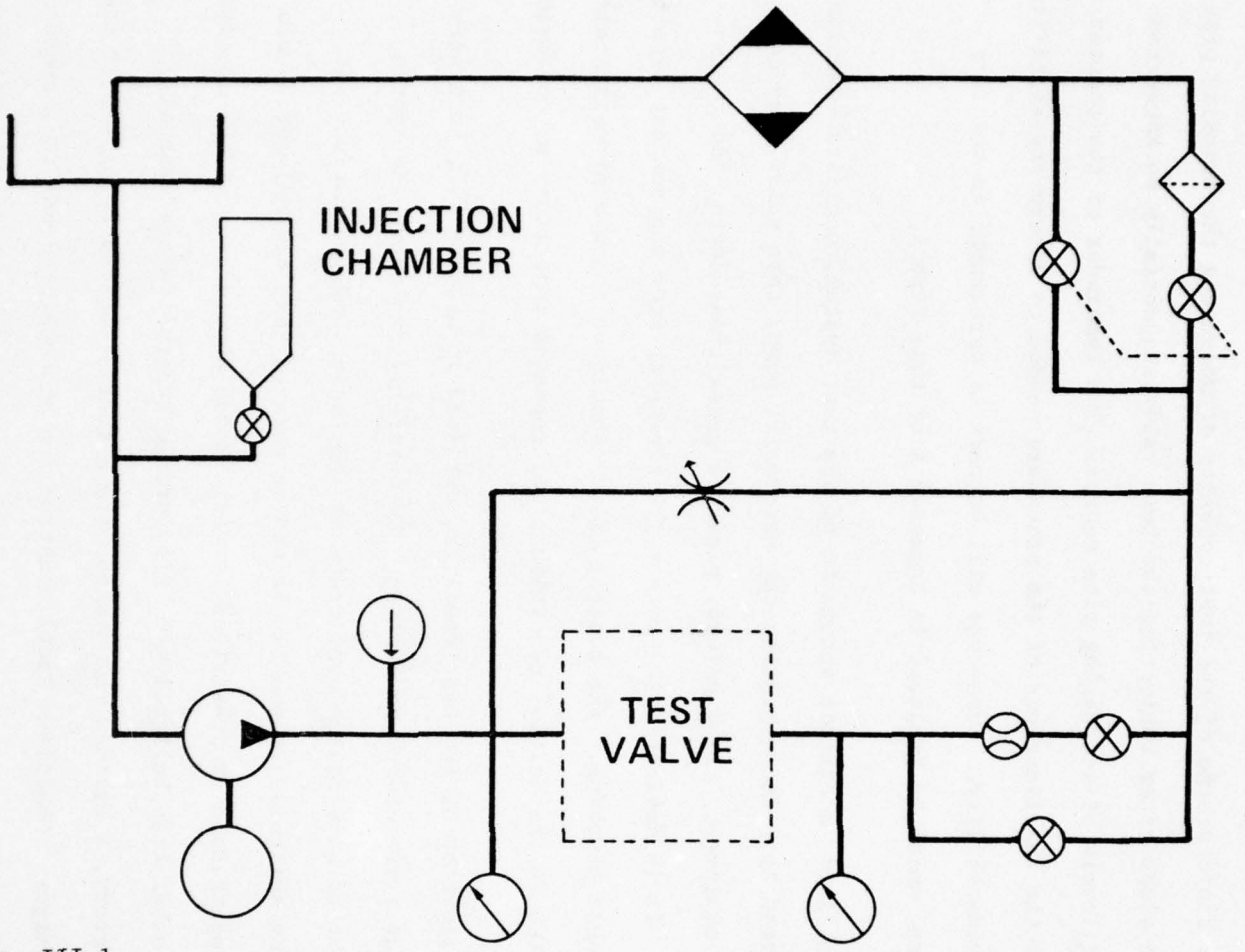


Fig. III-1

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level against that calculated from system volume estimates every time significant plumbing changes are made.

Section 9.1 describes the preliminary preparation. Note that the differential pressure across the valve, rather than the upstream pressure, is the parameter of interest. This measurement ensures that back pressure due to plumbing, oil filter, etc., does not affect the test. The static characteristic of the test valve is measured by the procedure outlines in section 9.2.1 through 9.2.3. Note that the flow rate should be kept continuously increasing between data points. This is necessary to eliminate errors due to valve hysteresis. Many valves exhibit different pressures for the same flow, depending on whether the flow was increasing or decreasing, and by always proceeding from low flow to high flow, it will be ensured that all data points are on the same characteristic. Section 9.2.5 describes the break-in procedure. Static characteristics are obtained in the beginning, middle and end of the break-in period.

Exposure to particulate contaminant is achieved by adding a known weight of ACFTD to the test system and circulating it for a given time. The size ranges indicated in section 9.3.2 are the same as for pump and motor contaminant sensitivity tests and the "cuts" are obtained by classifying ACFTD. Slurries can be prepared by mixing the prescribed weight of cut dust with 50-80 ml of oil and vigorously shaking the bottle. An ultrasonic shaker is effective in breaking down agglomerates.

The valve is subjected to particulate contaminant of a given size range for 30 minutes, the flow rate being half the specified valve. The system is cleaned by circulating fluid through the control filters and

only then is the static characteristic obtained. Experiments have indicated that characteristics measured in dirty fluid are not highly repeatable. Additionally, the flow meter may be damaged by exposure to contaminated fluid. Hence, all measurements should be made in clean fluid.

All the test information is contained in the pressure flow graphs. From these graphs, the cracking pressure after each exposure can be read and plotted against the cut-size. The slope of the characteristic is obtained by fitting a straight line to the data points. The slope after each exposure is plotted against the cut-size.

General degradation trends are hard to describe since there are great variations in design details from one valve to another. Due to increase of clearances and wear of sealing surface, most valves will exhibit lowering of cracking pressure with contaminant exposure. Changes in slope are harder to explain without dismantling and inspecting the internal parts.

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MERADCOM/OSU HYDRAULIC SYSTEM RELIABILITY PROGRAM.(U)
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CHAPTER III

INDUSTRIAL LIAISON

An important phase of the procedure development effort consisted of eliciting industry inputs on different aspects of the draft test procecure. The questionnaire which was distributed to the fluid power industry, together with the circulating list is included as Appendix B of this report. The remainder of this chapter summarizes the industry response and discusses its implications in terms of contaminant sensitivity testing and appraisal. Questions pertaining to subjects other than contaminant sensitivity testing are not discussed in this report. (See Appendix B.)

Question 1 was asked to elicit views on the definition of cracking pressure. A majority of the respondents (19 out of 33) indicated that it should be the pressure at which a specified percentage of the rated flow would be passed. Numerical values ranged from 1% to 5% of the rated flow. Most of the respondents who disagreed with a percentage rating desired an absolute value for the flow, ranging from "preceivable drops" to 0.5 gpm. Since valve flow ratings could range anywhere from 1 gpm (3.785 lpm) to 200 gpm (757 lpm) and more, the absolute rating would permit abnormally high flows for small valves and impractically small ones for large valves. Hence, the percentage rating rating has been retained in the test procedure.

Replying to a question asking about the validity of cracking and reaseal pressures as performance parameters, an overwhelming majority of respondents (29 out of 32) indicated in the affirmative. One negative response commented that cracking and reseal were not well defined, and

one abstained from responding. It is concluded that the industry as a whole recognizes the importance of cracking and reseal pressures. Hence the usage of the first in the contaminant sensitivity appraisal test appears acceptable to the whole industry.

A majority (19 out of 32) indicated that they felt that peak overshoot would be a good parameter for contaminant sensitivity tests (question 8). Most of the negative responses maintained that such a test would not be repeatable and some, including some of the positive responders, felt that contaminant may not have significant effect on overshoot. Since the current test procedure covers only static performance, the replies to this question are not pertinent. However, the response indicates a cognizance of the need for dynamic evaluation.

Responding to Question 10, a large majority (27 out of 32) indicated that hysteresis should be considered a valid parameter for relief valve testing. The negative respondents were concerned, for the most part, with repeatability of measurements. In order to keep the test as simple as possible, the proposed test procedure calls for cracking pressure measurements, rather than hysteresis. However, with a slight change in wording, and a little more experimental effort, hysteresis measurements can be made.

The control of test parameters is of vital importance to ensure repeatability of tests. Fourteen out of 32 indicated that pressure at inlet should be controlled to within 2%, while 10 indicated an accuracy of 3-5%. The corresponding figures for the outlet pressure were 13 and 10. Since the question was not clearly worded, quite a few abstained from responding.

Some responders indicated that control of tank port pressure and flow rate was critical to testing.

Question 12 elicited comments on selection of contaminant for valve testing. A majority (27 out of 32) considered ACFTD to be a satisfactory contaminant. A significant number (14) indicated that carbonyl iron should also be used, while six desired the use of aluminum. Since ACFTD is the internationally accepted contaminant for many fluid power components, its specification in the test procedure is considered appropriate. The test procedure would, of course, be applicable for other types of classified contaminants. Some companies indicated that mixtures should be used, and even though this can be justified on many grounds, simplicity and acceleration of test preclude such sophisticated arrangements.

In response to the question on background contamination level, there was no clear-cut majority. Eight responders indicated 1000 particles/ml greater than 10 μm as an acceptable leve, while 6 preferred 100 particles/ml greater than 10 μm. A large number either did not answer or referred to the use of 3-5 micrometer filter. Since 100 particles/ml of size 10μm or more is achievable with a number of commercially available filters, this valve has been incorporated in the test procedure. This valve is also used in pump and motor contaminant sensitivity testing.

Pump and motor tests subject the component to increasing sizes of contaminant. Responding to question no. 15, a majority (22 out of 32) indicated that a similar procedure would be satisfactory for relief valves. Some of the negative responders felt that fine contaminant would silt up

the valve, while others felt that the fine particles might damage the valve before the test was completed. One respondent suggested the use of a full distribution to make the test more realistic. The test procedure specifies that the contaminant size should be monotonically increased during the test.

Responding to the question as to the minimum starting size for contaminant, 13 out of 32 indicated a valve of 0-5μm, while another 15 specified 0-10μm. Since many valves are expected to operate in fluid having contaminants in the size range 0-5μm, the test procedure specified this as the starting range.

A majority of respondents (15 out of 32) indicated that 0-80μm would be a satisfactory terminal size for the test. Some felt that 0-40μm should be the maximum range while two indicated a range beyond 0-100μm. Since the number of particles per unit for a given gravimetric decreases as the size range is increased, the number of large particles, say greater than 100μm, is difficult to control and hence tests using 0-100μm and larger cuts may not be repeatable. An additional source of concern is the possibility of catastrophic failures due to particle jamming when large size ranges are used. Hence the test procedure limits the size range to 0-80μm.

Three out of thirty-two respondents indicated that 10 mg/l should be the minimum gravimetric level for valve tests, while one each specified 40 mg/l and 60 mg/l. Seven indicated that 100 mg/l should be the minimum valve while the majority (13) specified 300 mg/l. Part of the reason for the popularity of the last figure could be attributed to its use in pump

and motor testing. Lower gravimetric levels can generally be used if:

(1) instrumentation accuracy is good enough to pick up change in performance parameters,

(2) random errors are kept small.

At larger particle size ranges, lower gravimetrics imply less number of particles and hence more data scatter. A majority of respondents indicated that 300 mg/l was the maximum gravimetric level for conducting valve tests. A slightly smaller number, 8, suggested 600 mg/l. In order to maintain uniformity with pump and motor test procedures, the same gravimetric level (300 mg/litre) has been specified. Larger valves might degrade the valves so rapidly as to render the last injection ineffective.

A very narrow majority (13 to 12, with 7 abstentions) are in favor of accelerated contaminant tests. Most of the negative respondents, and some of the positive respondents, feel that an accelerated test is unrealistic, at least in so far as life estimations are concerned. Nevertheless, such a test is still considered valid for relative appraisal of alternative designs. Extensive field data acquisition and laboratory tests are needed to build up a reliable data base before definitive remarks on the validity of accelerated tests can be made.

In conclusion, the test procedure reflects the concensus of the fluid power industry and can form the basis for a national voluntary standard. Some of the questions raised by the respondents can only be answered by additional laboratory testing, and the correlation between laboratory and field degradation will have to wait until sufficient field data are collected.

CHAPTER IV

EXPERIMENTAL RESULTS

Contaminant sensitivity tests were conducted as per the test procedure outlined in Appendix A on two relief valves. The results of the tests are summarized below separately.

OSU Valve ID: RV-25

Type of Valve:	Two Stage Relief
Specified Flow:	30 gpm (113.6 lpm)
Specified Pressure:	1500 psi (10.35 MPa)
Test Fluid:	Mil-L-2104C, Class 10
Operating Temperature:	120°F (49°C)

Exposure to particulate contaminant was done at a flow rate of 56.8 lpm, one-half the specified flow. The cracking pressure corresponded to a flow rate of 5.68 lpm. Table IV-1 summarizes the pressure flow characteristics of the valve, before break-in (PQ_I) after 10 minutes in clean fluid (PQ_{II}), and after 20 minutes in clean fluid (PQ_S). The latter two are sufficiently close to infer that the valve has reached stable operation. Also contained in Table IV-1 are the data for the characteristic after injection of contaminant, starting with 0-5 μm and terminating in 0-80 μm. The headings for the data sets are self-explanatory. Note that all pressure-flow characteristics were obtained by increasing the flow monotonically from below 5.68 lpm to 56.8 lpm. This procedure is necessary to ensure that hysteresis does not introduce errors in the data. Figures IV-1 through IV-12 plot the pressure flow characteristics, while Fig. IV-13 and IV-14 plot the cracking pressure average slope against contaminant size.

TABLE IV-1

Degradation of Relief Valve RV-25

Graph PQ_I

Flowrate (lpm)	ΔP (MPa)
5.68	10.04
11.40	10.07
17.00	10.10
22.70	10.20
28.40	10.26
34.10	10.33
39.70	10.39
45.40	10.41
51.10	10.46
56.80	10.49

Graph PQ_{II}

Flowrate (lpm)	ΔP (MPa)
5.68	9.87
11.40	9.93
17.00	10.04
22.70	10.14
28.40	10.20
34.10	10.26
39.70	10.32
45.40	10.35
51.10	10.45
56.80	10.55

Graph PQ_S

Flowrate (lpm)	ΔP (MPa)
5.68	9.91
11.40	10.05
17.00	10.11
22.70	10.21
28.40	10.24
34.10	10.30
39.70	10.37
45.40	10.44
51.10	10.49
56.80	10.52

Graph PQ_{05}

Flowrate (lpm)	ΔP (MPa)
5.68	9.89
11.40	9.96
17.00	10.06
22.70	10.12
28.40	10.19
34.10	10.25
39.70	10.31
45.40	10.34
51.10	10.41
56.80	10.46

Graph PQ_{010}

Flowrate (lpm)	ΔP (MPa)
5.68	9.82
11.40	9.92
17.00	9.95
22.70	10.06
28.40	10.15
34.10	10.21
39.70	10.24
45.40	10.30
51.10	10.37
56.80	10.39

Graph PQ_{020}

Flowrate (lpm)	ΔP (MPa)
5.68	9.83
11.40	9.89
17.00	9.92
22.70	9.96
28.40	9.99
34.10	9.98
39.70	10.00
45.40	10.04
51.10	10.06
56.80	10.08

Graph PQ_{030}

Flowrate (lpm)	ΔP (MPa)
5.68	10.10
11.40	10.15
17.00	9.84
22.70	9.81
28.40	9.75
34.10	9.70
39.70	9.66
45.40	9.62
51.10	9.61
56.80	9.56

Graph PQ_{040}

Flowrate (lpm)	ΔP (MPa)
5.68	10.56
11.40	10.04
17.00	9.73
22.70	9.60
18.40	9.52
34.10	9.41
39.70	9.34
45.40	9.30
51.10	9.26
56.80	8.94

Graph PQ_{050}

Flowrate (lpm)	ΔP (MPa)
5.68	9.91
11.40	9.88
17.00	9.84
22.70	9.81
28.40	9.77
34.10	9.74
39.70	9.74
45.40	9.62
51.10	9.60
56.80	9.64

Graph PQ_{060}

Flowrate (lpm)	ΔP (MPa)
5.68	9.68
11.40	9.82
17.00	9.89
22.70	9.89
28.40	10.07
34.10	10.24
39.70	10.38
45.40	10.59
51.10	10.83
56.80	10.89

Graph PQ_{070}

Flowrate (lpm)	ΔP (MPa)
5.68	9.39
11.40	9.81
17.00	10.09
22.70	10.30
28.40	10.92
34.10	11.50
39.70	12.32
45.40	12.91
51.10	13.60
56.80	13.94

Graph PQ_{080}

Flowrate (lpm)	ΔP (MPa)
5.68	9.35
11.40	9.32
17.00	9.35
22.70	9.38
28.40	9.31
34.10	9.38
39.70	9.47
45.40	9.57
51.10	9.68
56.80	9.77

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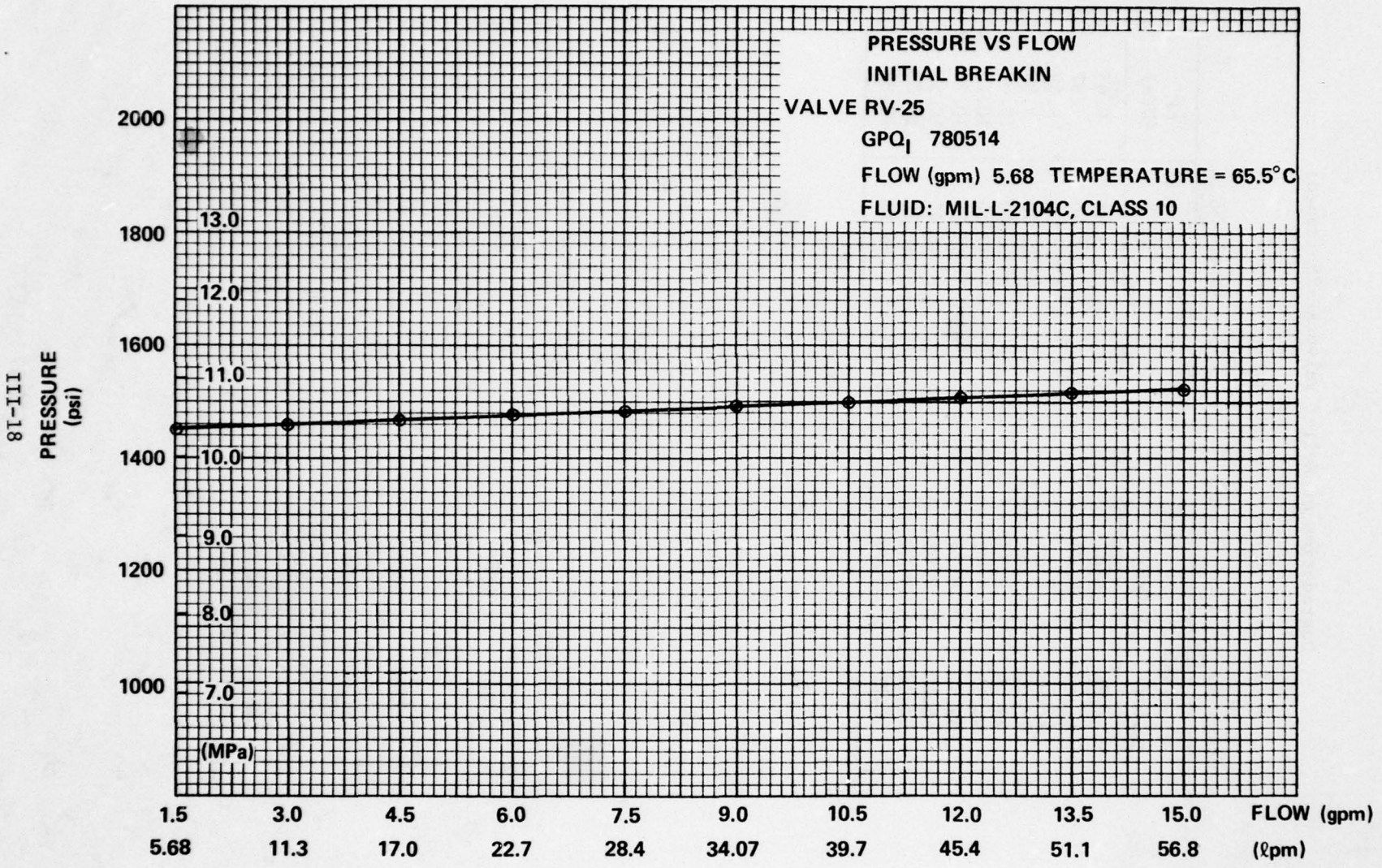


Fig. IV-1

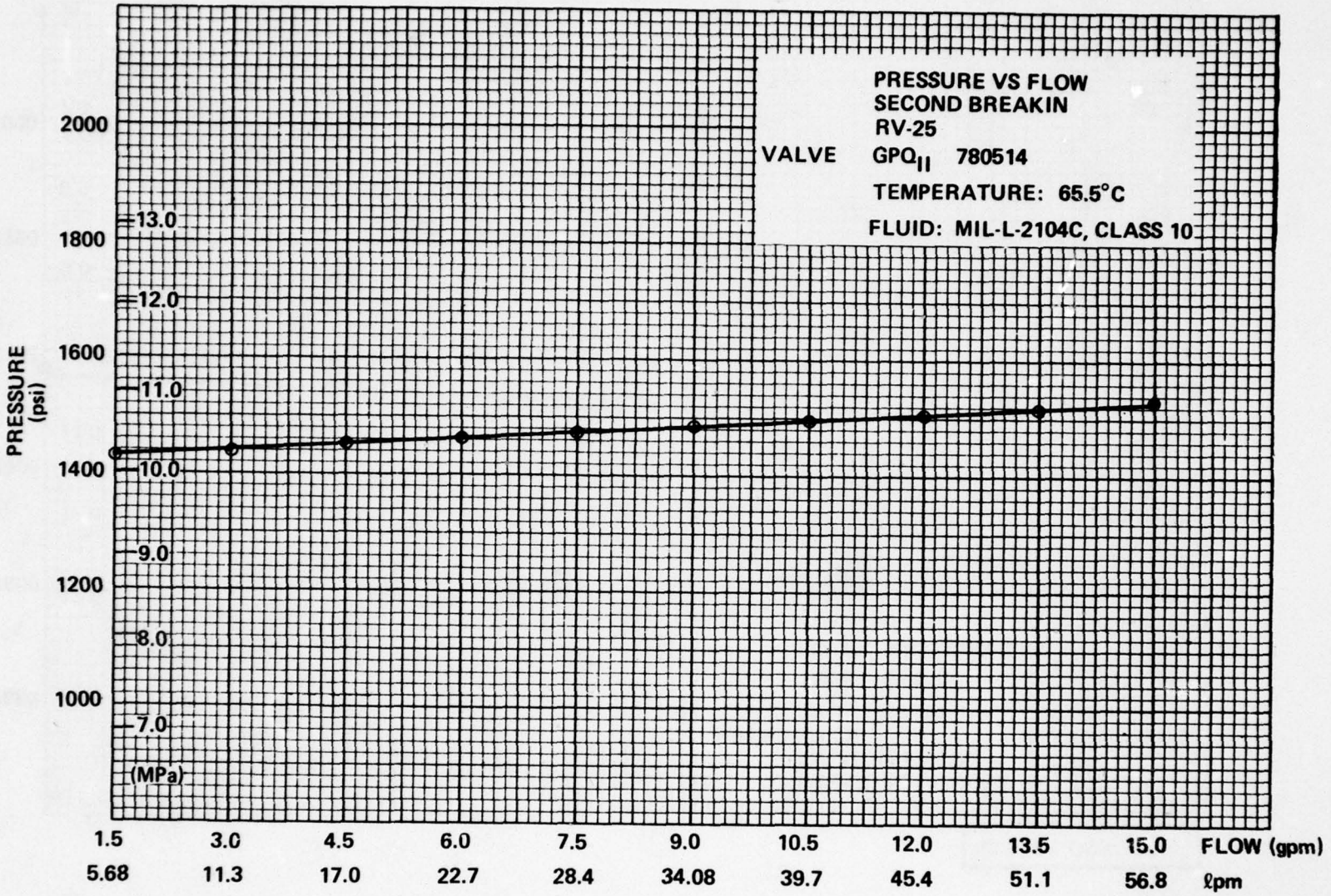


Fig. IV-2

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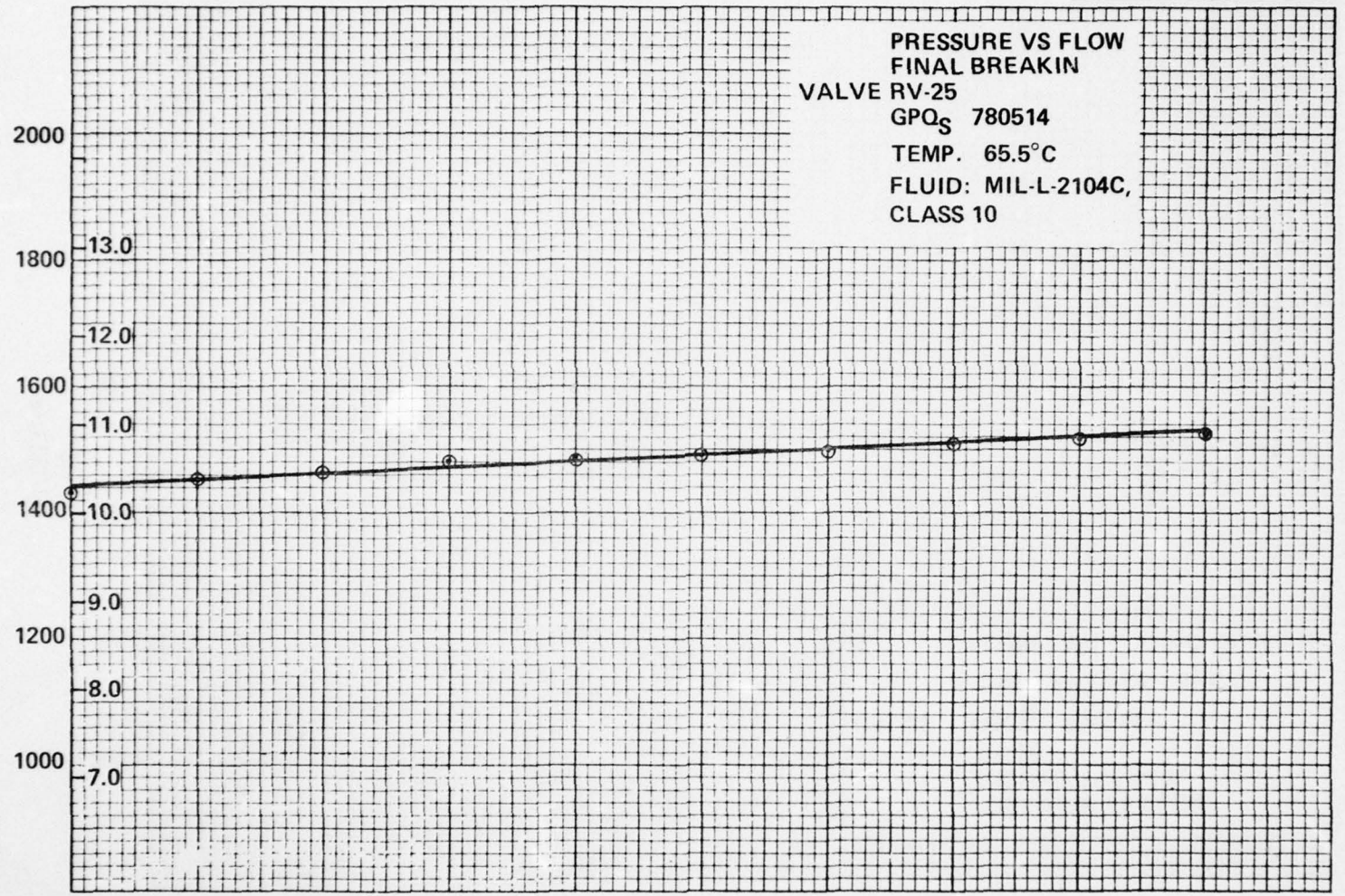


Fig. IV-3

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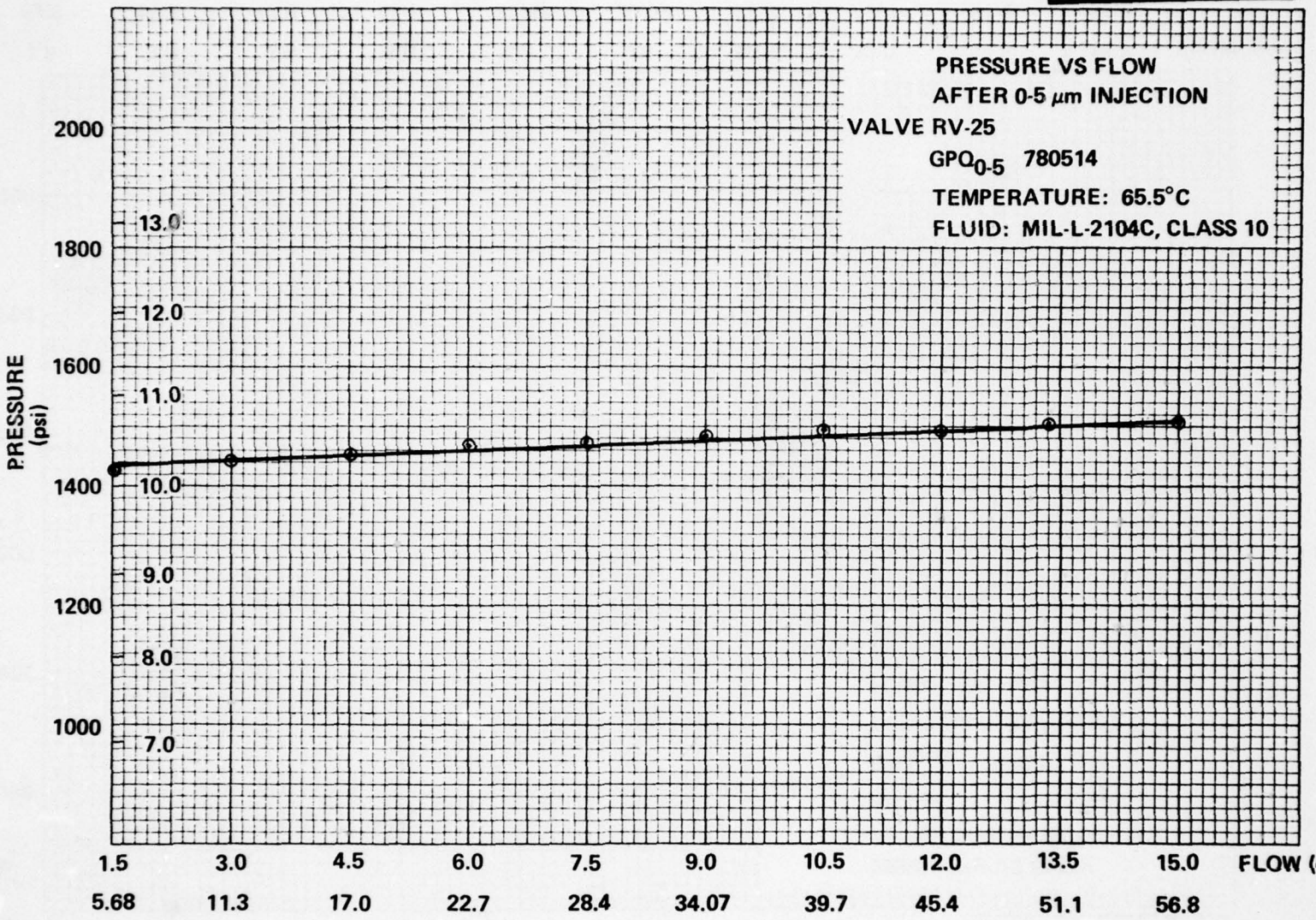


Fig. IV-4

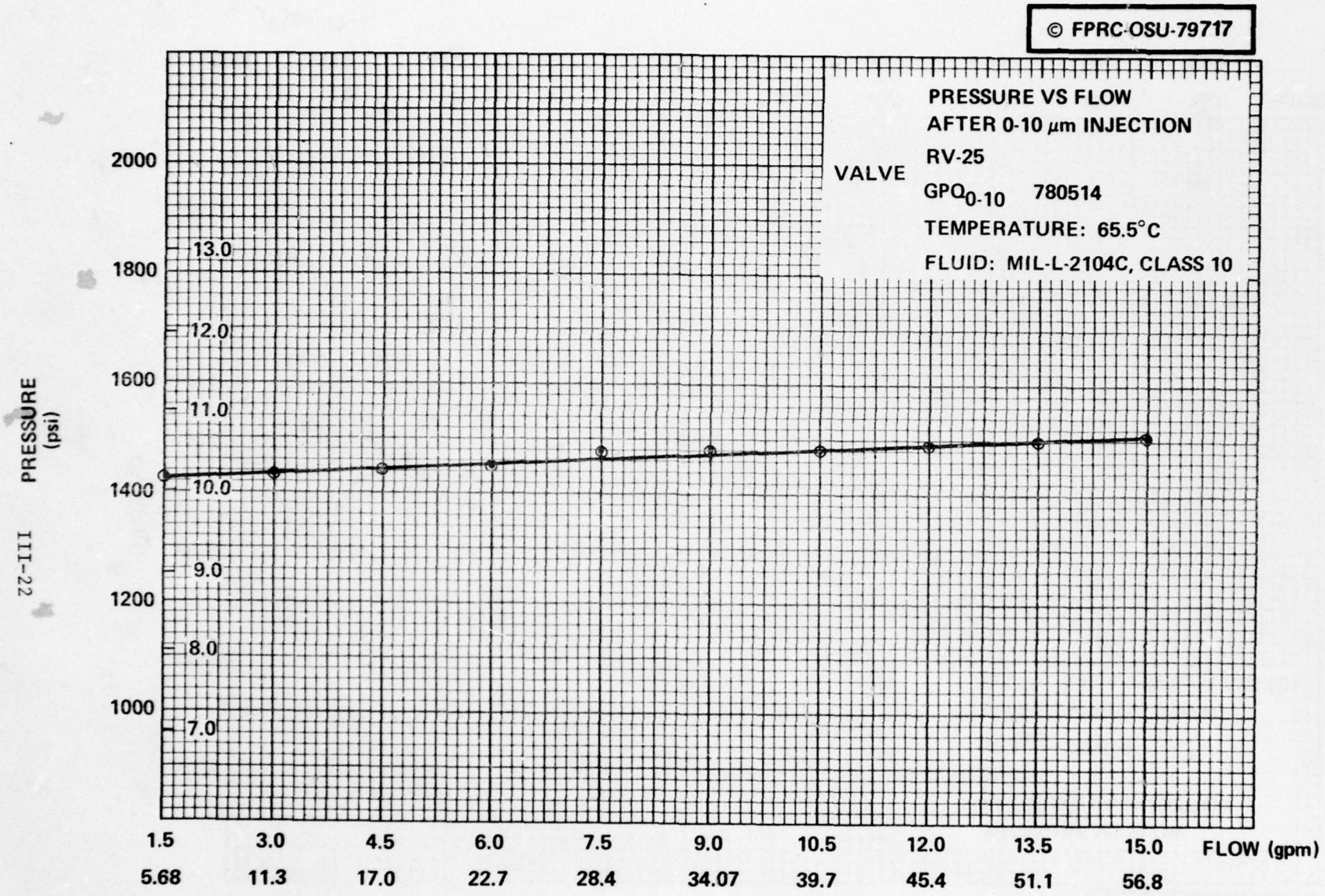

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PRESSURE VS FLOW
AFTER 0-10 μm INJECTION
VALVE RV-25
GPO0-10 780514
TEMPERATURE: 65.5°C
FLUID: MIL-L-2104C, CLASS 10
PRESSURE (psi)
2000
1800
1600
1400
1200
1000
13.0
12.0
11.0
10.0
9.0
8.0
7.0
1.5 3.0 4.5 6.0 7.5 9.0 10.5 12.0 13.5 15.0 FLOW (gpm)
5.68 11.3 17.0 22.7 28.4 34.07 39.7 45.4 51.1 56.8

Fig. IV-5

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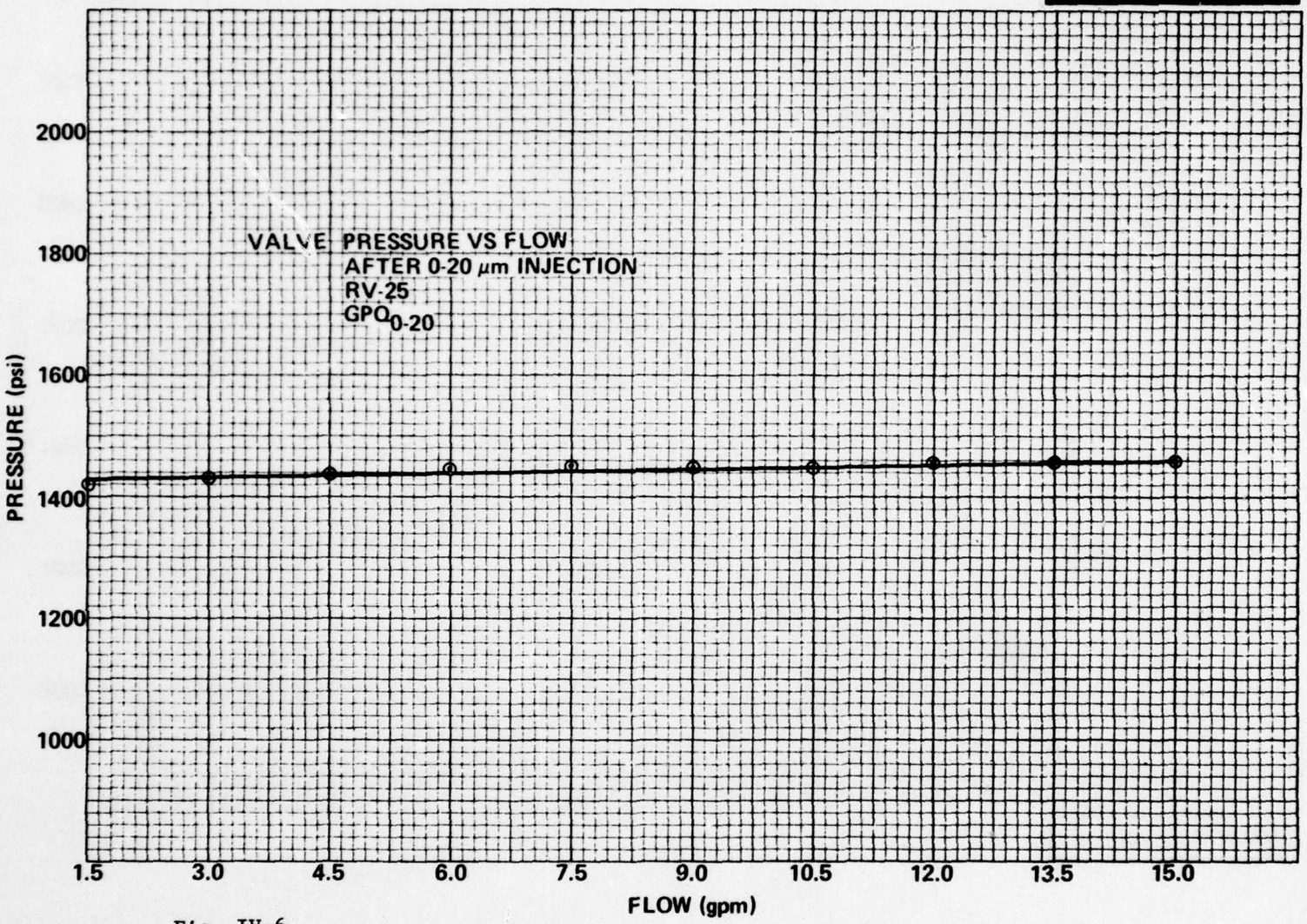


Fig. IV-6

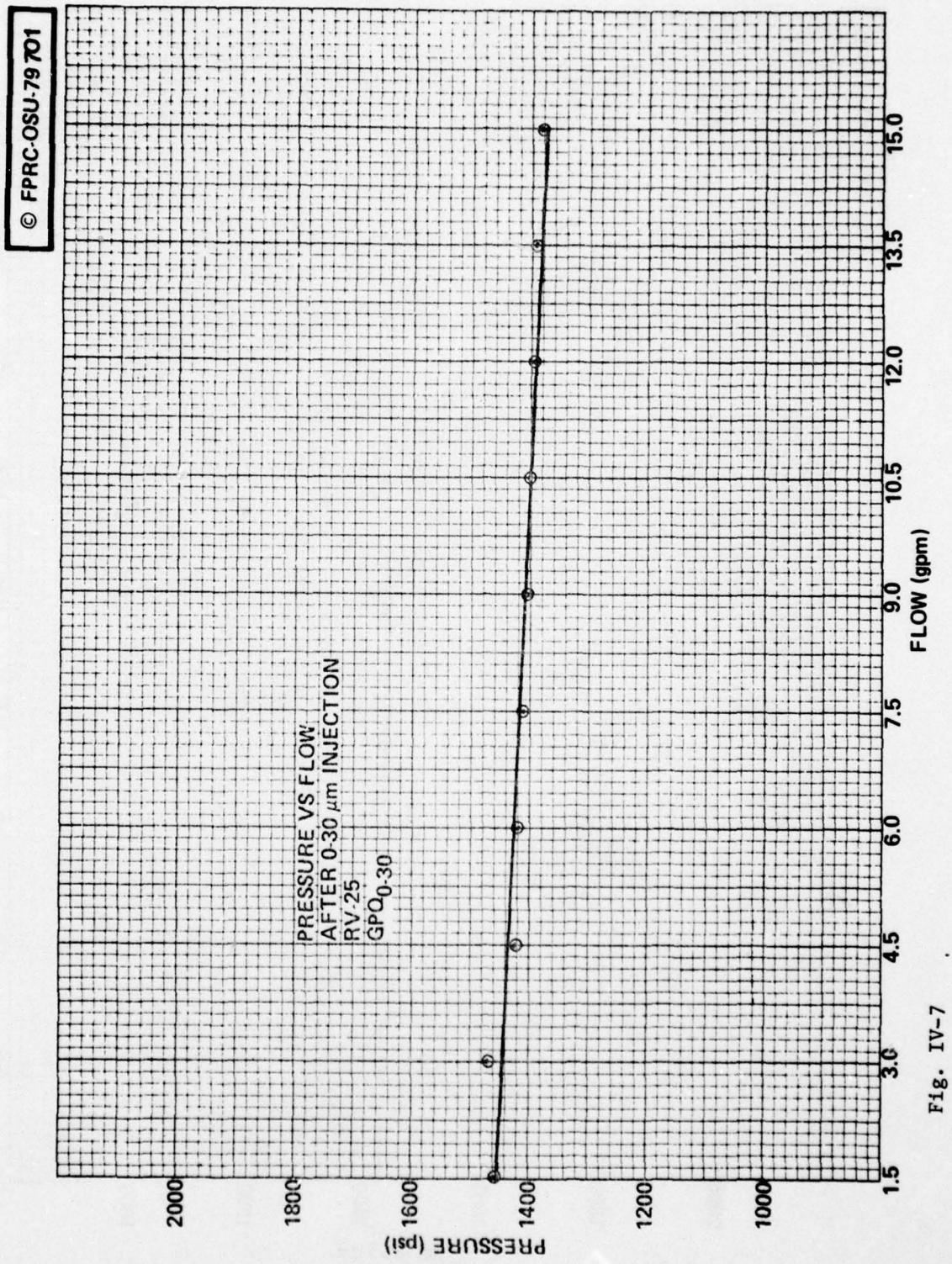

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PRESSURE VS FLOW
AFTER 0-30 μm INJECTION
RV-25
GPO0-30
PRESSURE (psi)
2000
1800
1600
1400
1200
1000
1.5
3.0
4.5
6.0
7.5
9.0
10.5
12.0
13.5
15.0
FLOW (gpm)

Fig. IV-7

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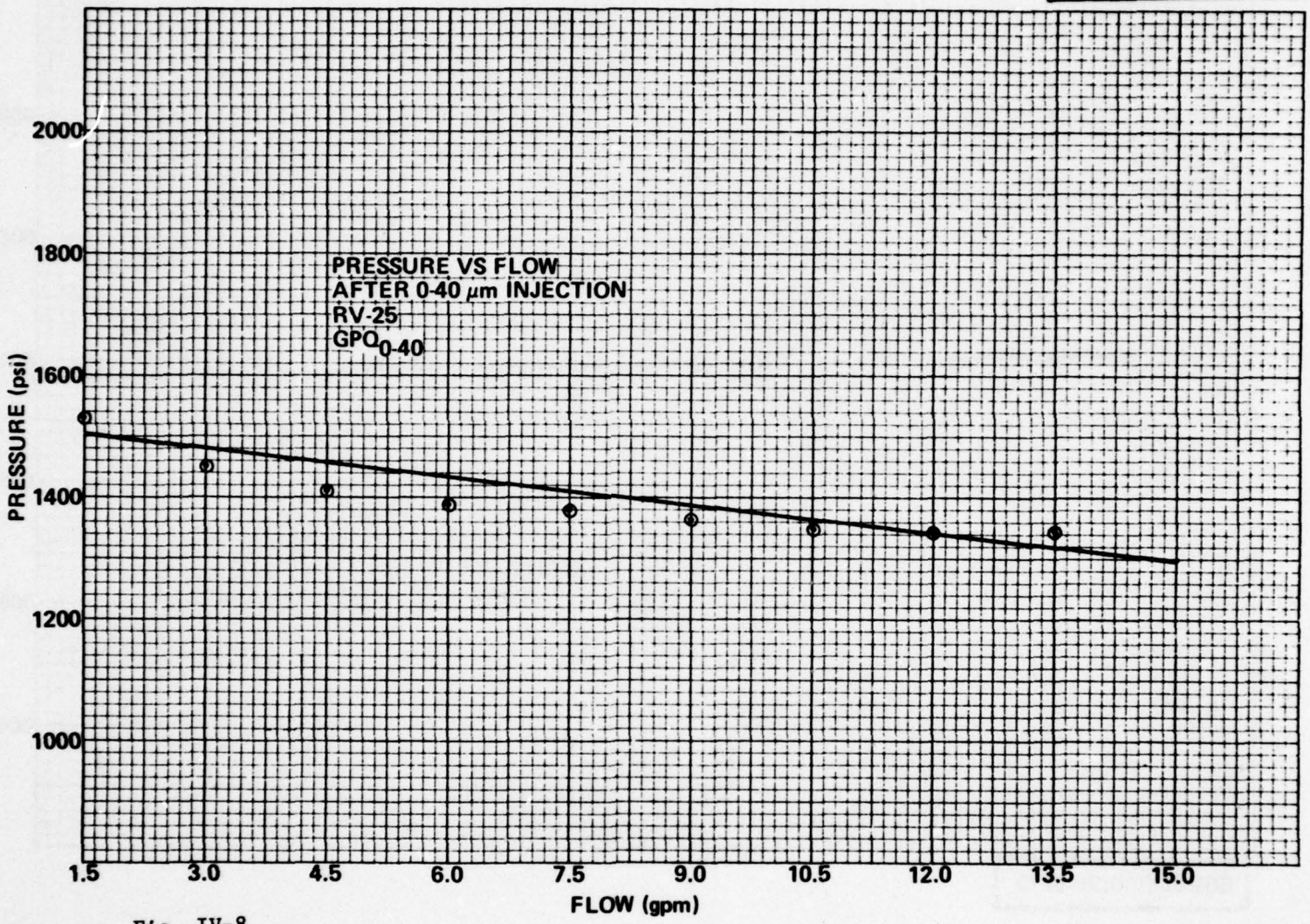


Fig. IV-8

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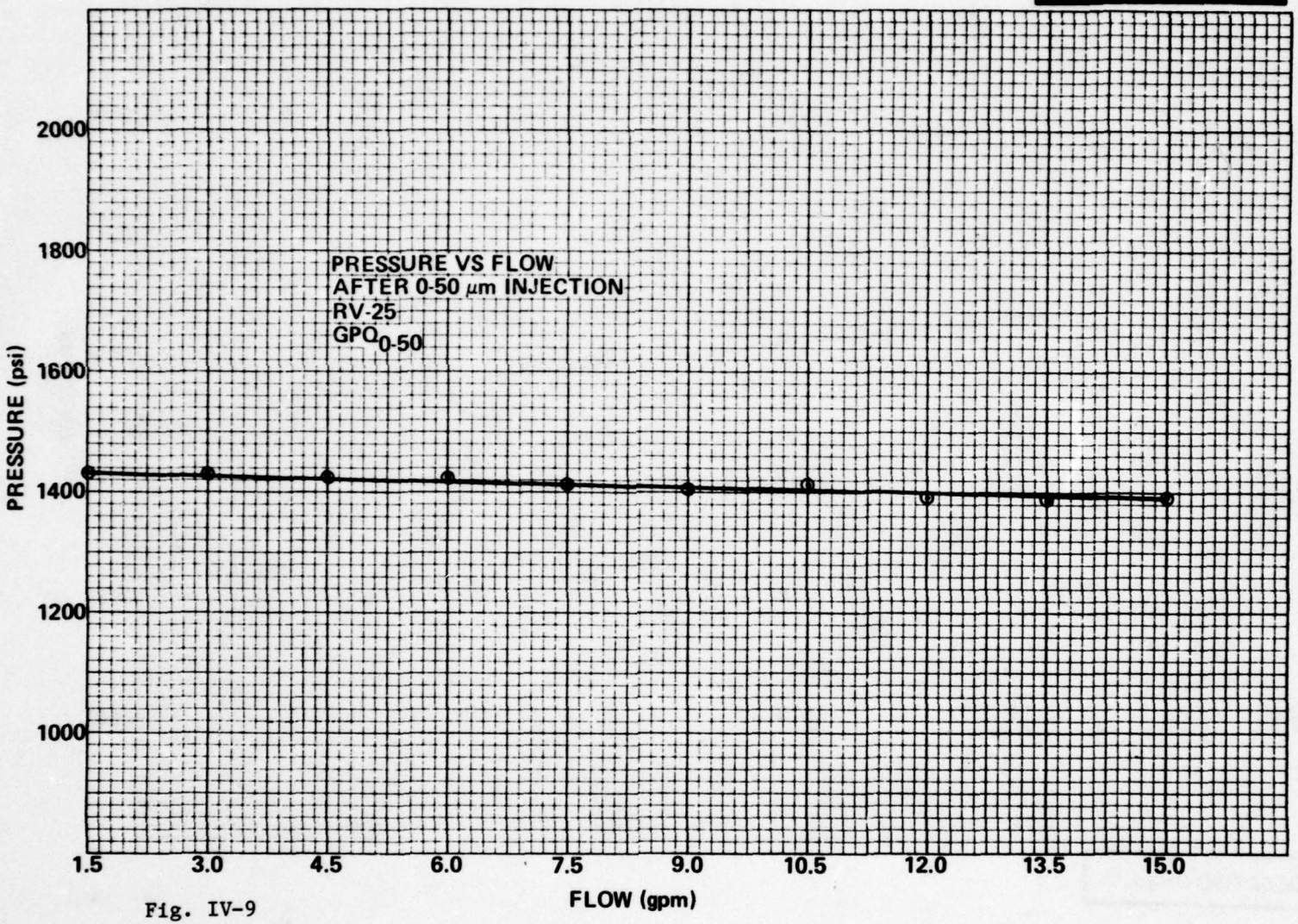


Fig. IV-9

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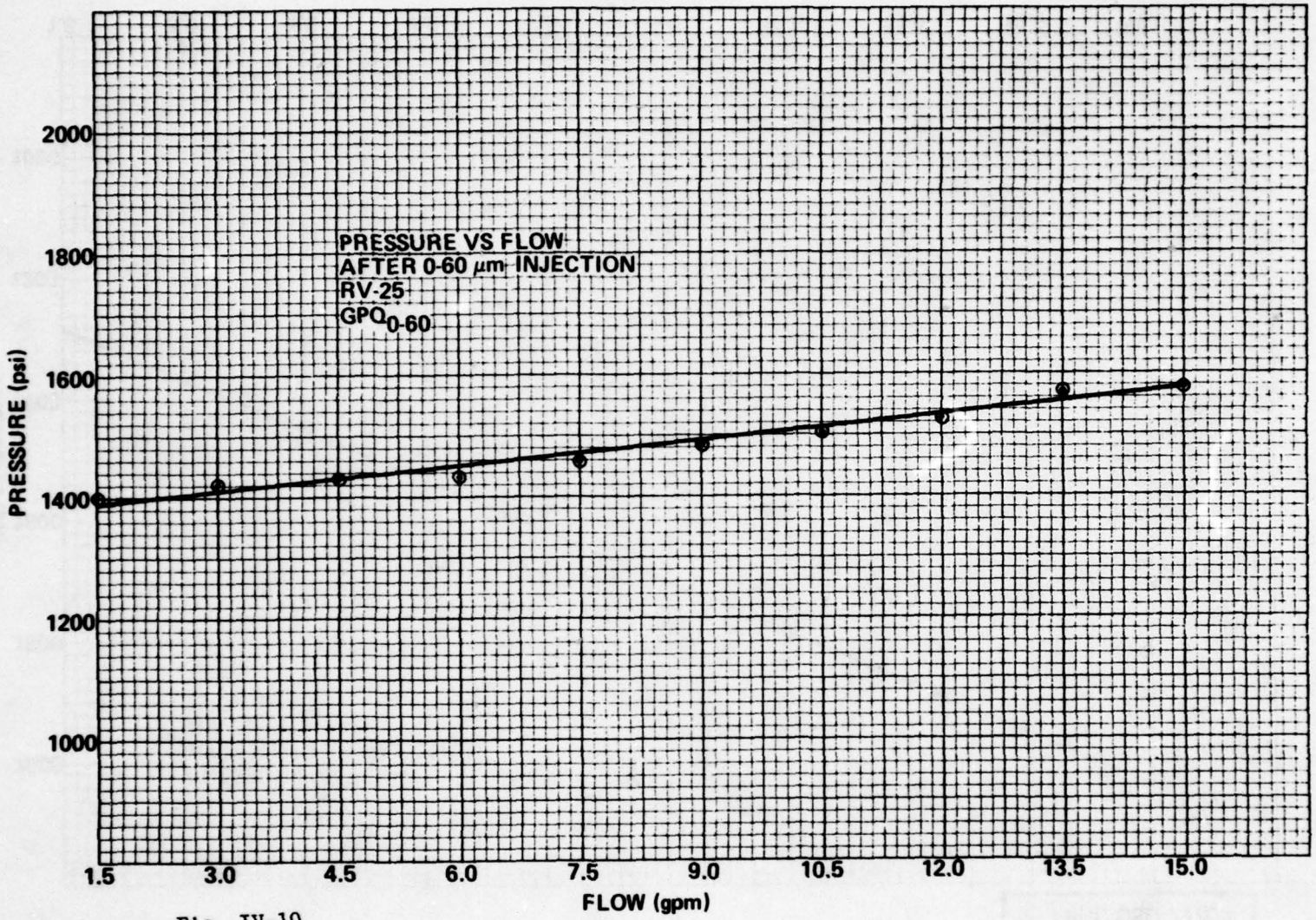


Fig. IV-10

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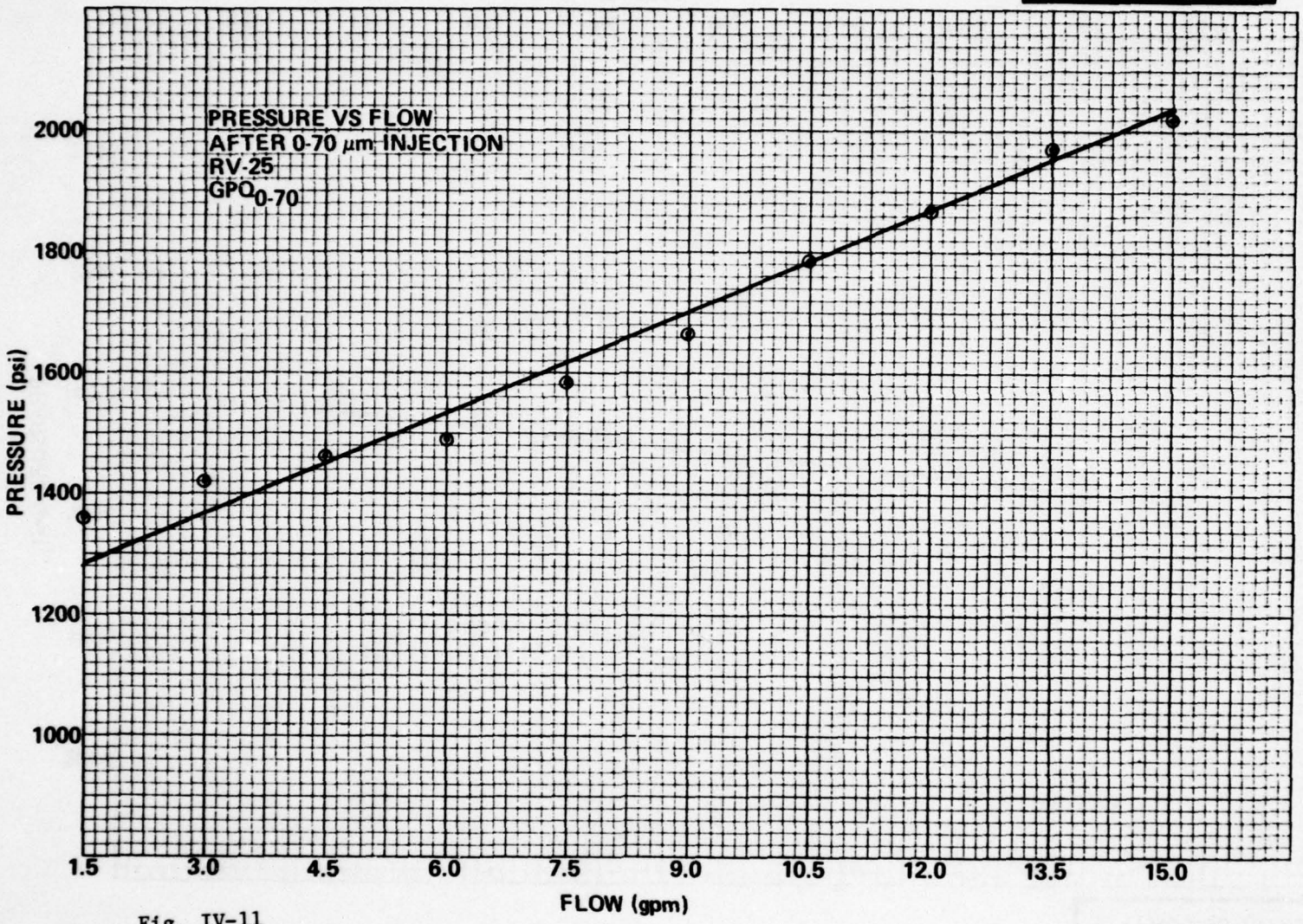


Fig. IV-11

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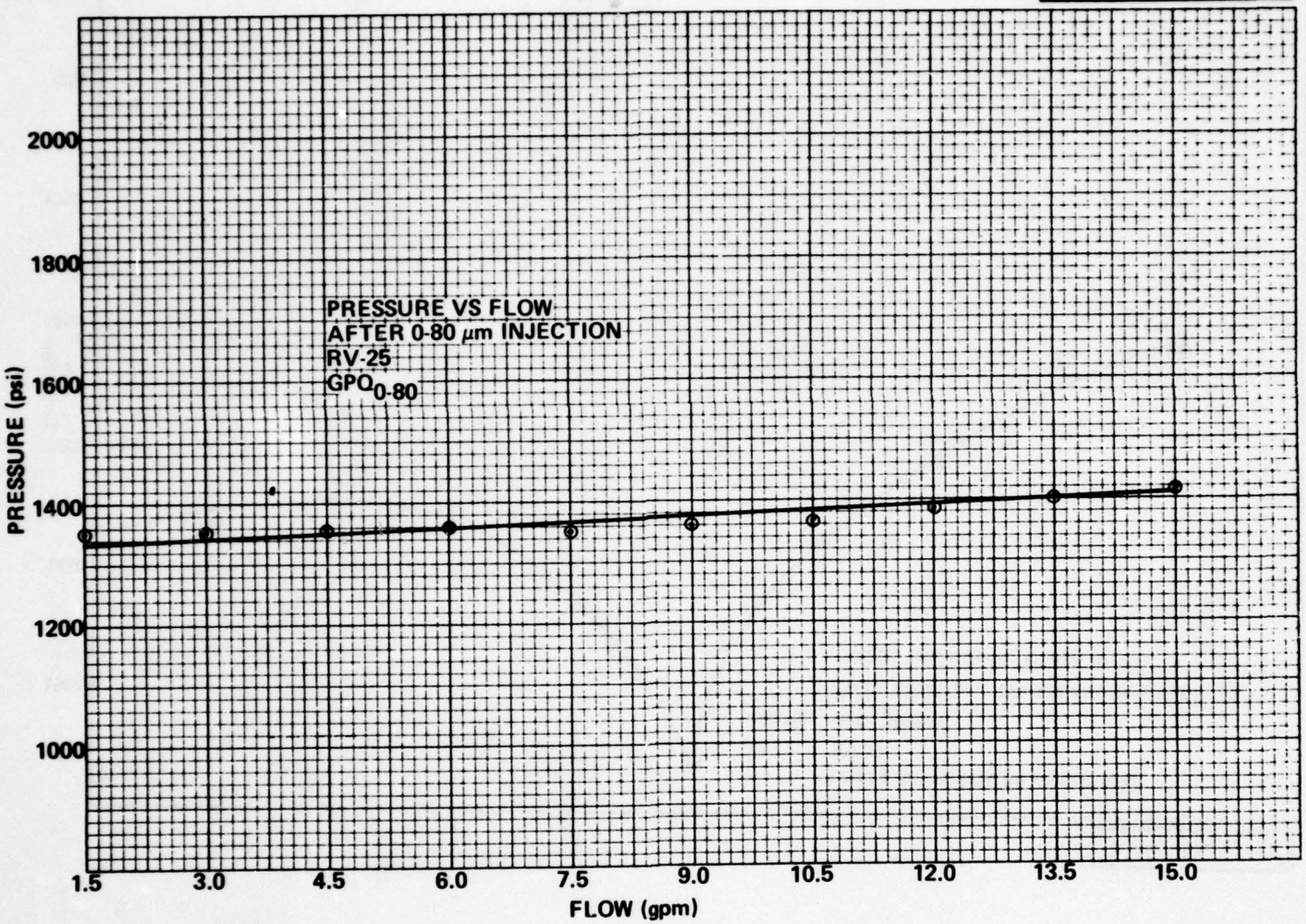


Fig. IV-12

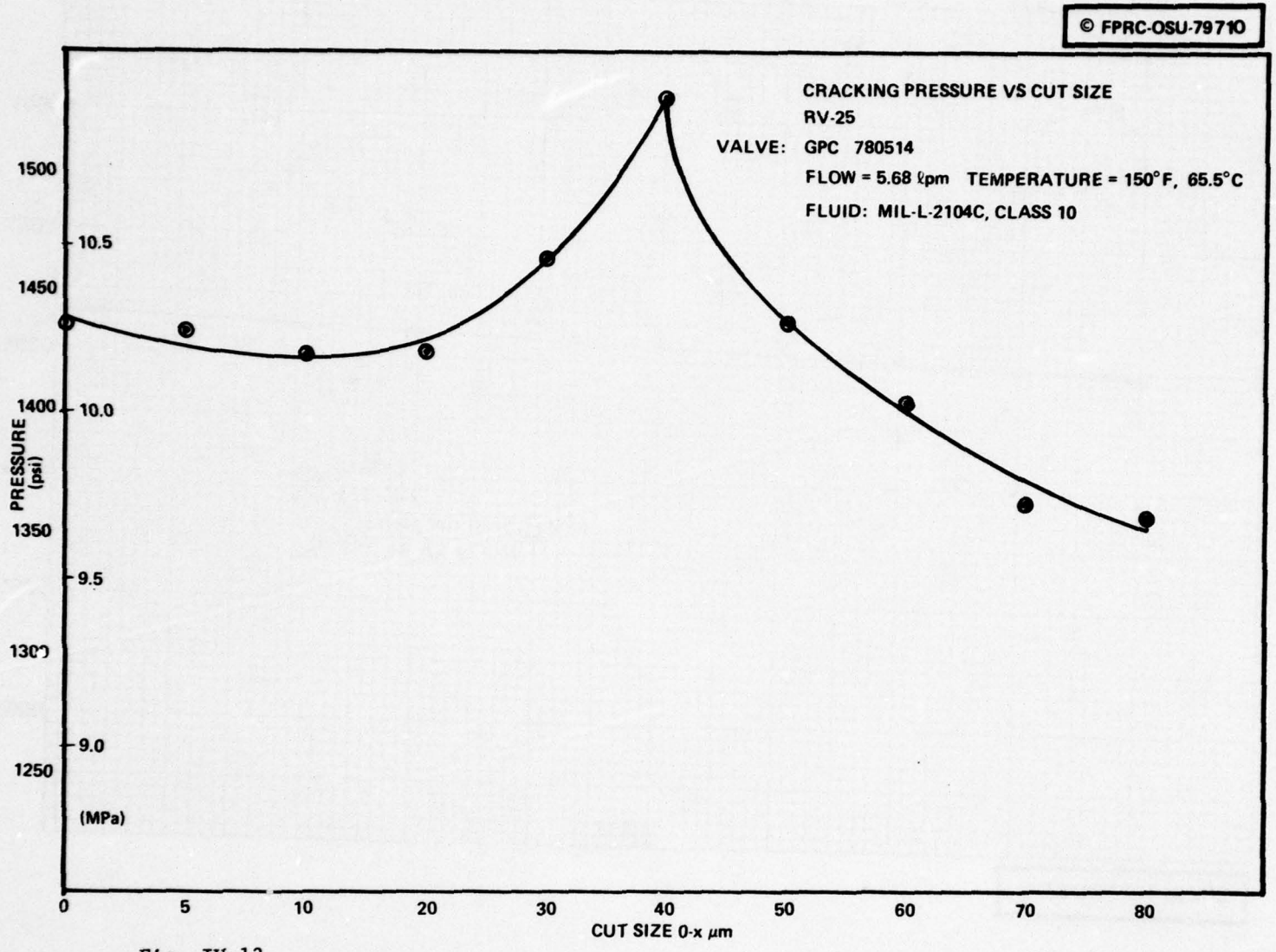

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CRACKING PRESSURE VS CUT SIZE
RV-25
VALVE: GPC 780514
FLOW = 5.68 ℓpm TEMPERATURE = 150°F, 65.5°C
FLUID: MIL-L-2104C, CLASS 10
PRESSURE (psi)
1500
1450
1400
1350
1300
1250
10.5
10.0
9.5
9.0
(MPa)
0
5
10
20
30
40
50
60
70
80
CUT SIZE 0-x μm

Fig. IV-13

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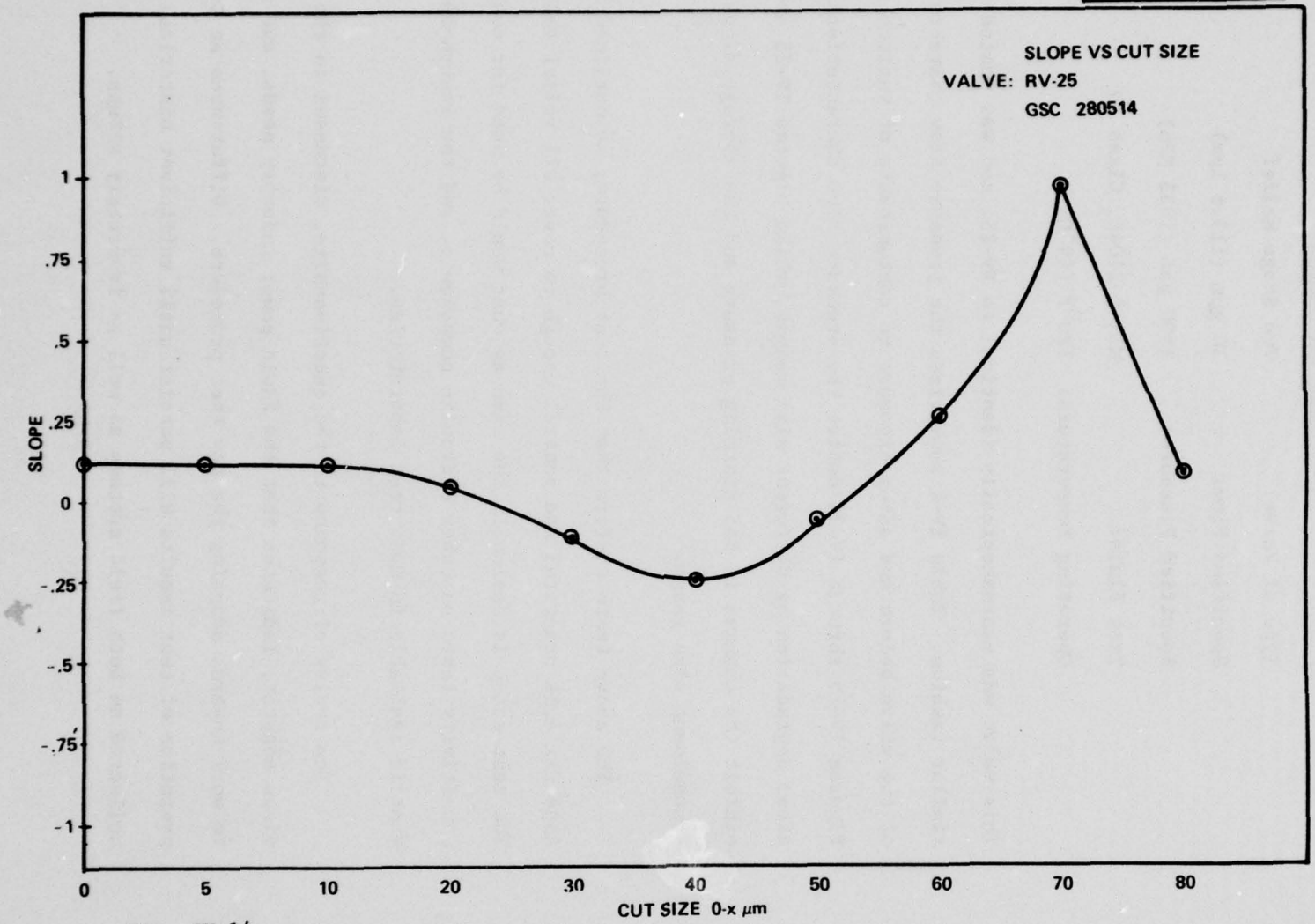


Fig. IV-14

OSU Valve ID: RV-26

Type of Valve:	Two Stage Relief
Specified Flow:	30 gpm (113.6 lpm)
Specified Pressure:	1500 psi (1035 MPa)
Test Fluid:	Mil-L-2104C, Class 10
Operating Temperature:	120°F (49^{9}C)

This valve was macroscopically identical to RV-25, and was tested in a similar fashion. Table IV-2 summarizes the pressure flow characteristics of the valve before and after exposure to contaminants of various sizes. Figures IV-15 through IV-24 depict the pressure flow characteristics after degradation by different size ranges, while Figures IV-25 and IV-26 exhibit the changes in the cracking pressure and the average slope vs the contaminant size range.

The above tests confirm that the test procedure, as outlined in Appendix A,is practical and general enough to cover all relief valves. The test setup is basically the same as what would be used for contaminant sensitivity tests on other hydraulic components, and the instrumentation, what is available in most test laboratories.

The review of response to the questionnaire, discussed in the previous chapter, indicates that the fluid power industry needs, and is willing to work towards adopting the new test procedure. Differences as to interpretation of test results will persist until sufficient historical data is collected on both field systems as well as laboratory setups.

Table IV-2

Degradation of Relief Valve RV-26

Graph PQ_I

Flowrate (lpm)	ΔP (MPa)
5.68	10.21
11.40	10.29
17.00	10.36
22.70	10.44
28.40	10.01
34.10	9.98
39.70	10.00
45.40	10.04
51.10	10.14
56.60	10.13

Graph PQ_{II}

Flowrate (lpm)	ΔP (MPa)
5.68	9.80
11.40	9.88
17.00	9.95
22.70	9.97
28.40	10.00
34.10	10.03
39.70	10.06
45.40	10.05
51.10	10.04
56.60	10.07

Graph PQ_S

Flowrate (lpm)	ΔP (MPa)
5.68	9.71
11.40	9.81
17.00	9.84
22.70	9.91
28.40	9.97
34.10	10.00
39.70	9.99
45.40	10.01
51.10	10.00
56.60	10.04

Graph PQ_{05}

Flowrate (lpm)	ΔP (MPa)
5.68	10.75
11.40	10.79
17.00	10.82
22.70	10.89
28.40	10.99
34.10	10.98
39.70	11.01
45.40	10.99
51.10	11.02
56.60	11.05

Graph PQ_{010}

Flowrate (lpm)	ΔP (MPa)
5.68	10.75
11.40	10.84
17.00	10.84
22.70	10.87
28.40	10.97
34.10	10.92
39.70	10.95
45.40	10.95
51.10	10.94
56.80	10.93

Graph PQ_{020}

Flowrate (lpm)	ΔP (MPa)
5.68	10.42
11.40	10.46
17.00	10.45
22.70	10.51
28.40	10.53
34.10	10.53
39.70	10.46
45.40	10.45
51.10	10.39
56.80	10.35

Graph PQ_{030}

Flowrate (lpm)	ΔP (MPa)
5.68	10.52
11.40	10.55
17.00	10.48
22.70	10.40
28.40	10.33
34.10	10.21
39.70	10.11
45.40	10.03
51.10	9.92
56.80	9.91

Graph PQ_{040}

Flowrate (lpm)	ΔP (MPa)
5.68	10.55
11.40	10.49
17.00	10.35
22.70	10.28
28.40	10.14
34.10	9.92
39.70	9.81
45.40	9.71
51.10	9.60
56.80	9.52

Graph PQ_{050}

Flowrate (lpm)	ΔP (MPa)
5.68	10.53
11.40	10.43
17.00	10.28
22.70	10.13
28.40	9.96
34.10	9.81
39.70	9.73
45.40	9.66
51.10	9.57
56.80	9.52

Graph PQ_{060}

Flowrate (lpm)	ΔP (MPa)
5.68	9.90
11.40	9.76
17.00	9.79
22.70	9.85
28.40	9.77
34.10	9.73
39.70	9.65
45.40	9.64
51.10	9.60
56.80	9.66

Graph PQ_{070}

Flowrate (lpm)	ΔP (MPa)
5.68	9.79
11.40	9.97
17.00	9.99
22.70	10.10
28.40	10.26
34.10	10.32
39.70	10.41
45.40	10.55
51.10	10.75
56.80	10.84

Graph PQ_{080}

Flowrate (lpm)	ΔP (MPa)
5.68	9.95
11.40	10.00
17.00	9.92
22.70	10.45
28.40	10.61
34.10	11.06
39.70	11.46
45.40	11.95
51.10	12.49
56.80	12.83

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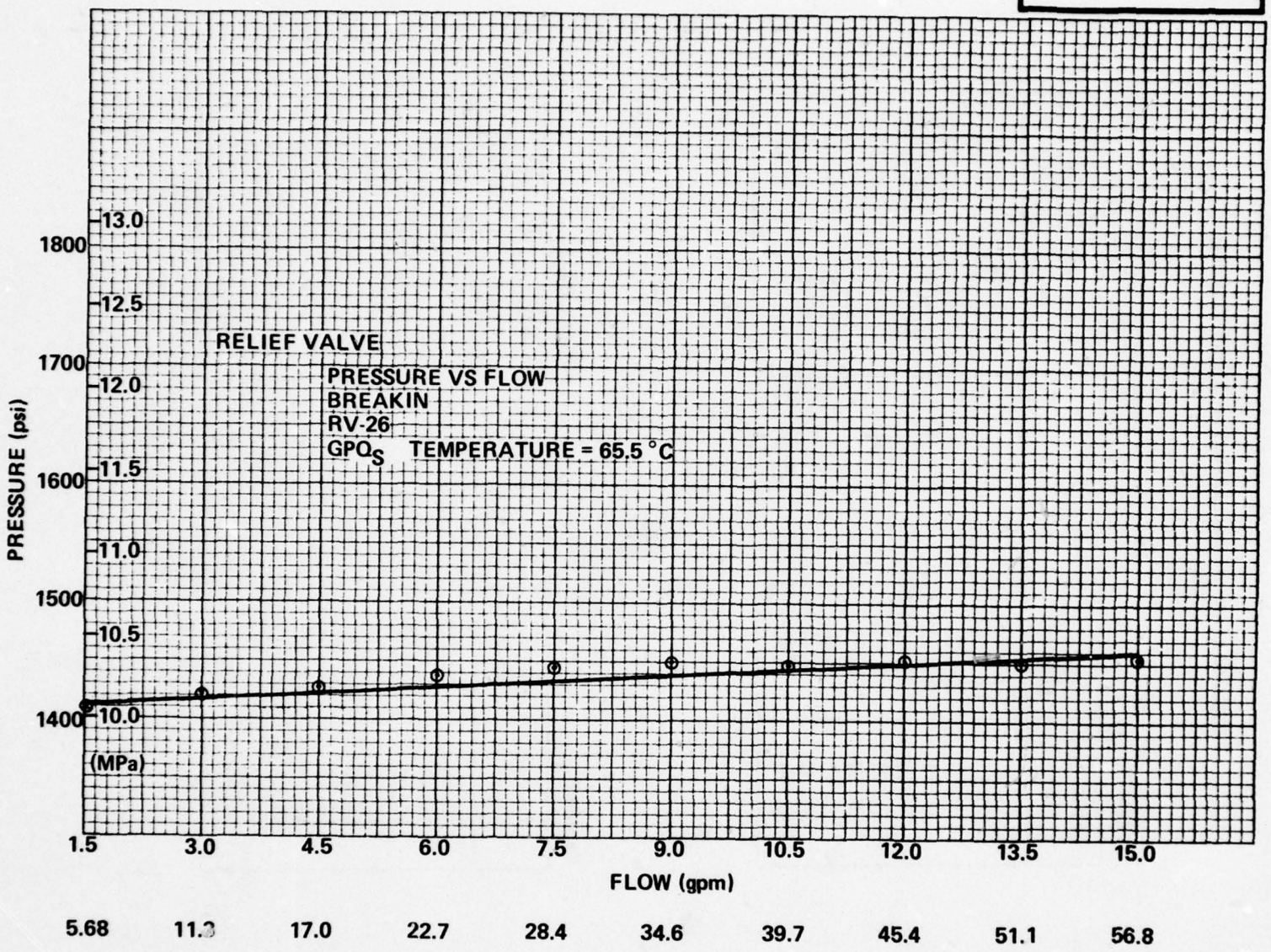


Fig. IV-15

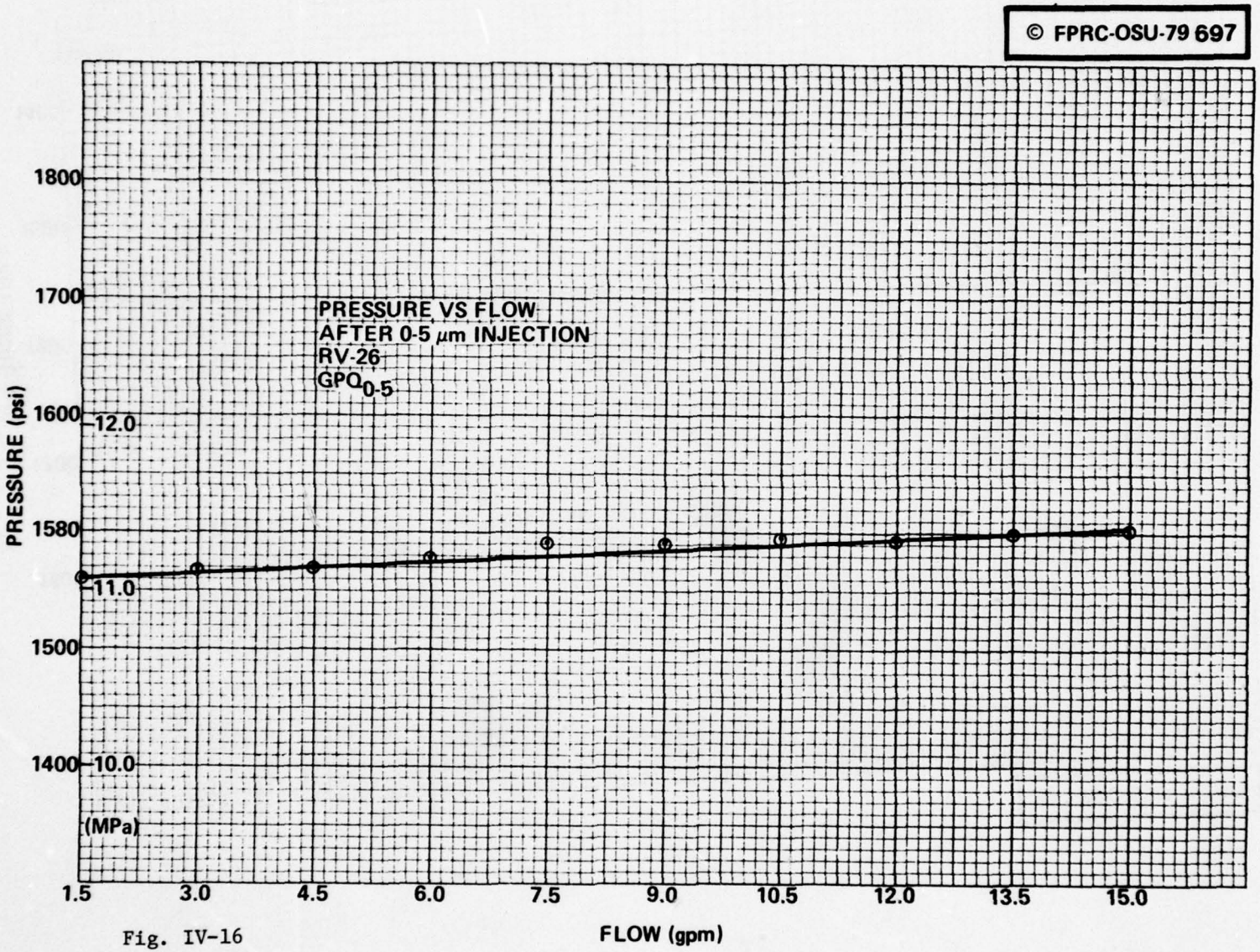

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PRESSURE VS FLOW
AFTER 0-5 μm INJECTION
RV-26
GPQ0-5
PRESSURE (psi)
1800
1700
1600
1580
1500
1400
12.0
11.0
10.0
(MPa)
1.5
3.0
4.5
6.0
7.5
9.0
10.5
12.0
13.5
15.0
FLOW (gpm)

Fig. IV-16

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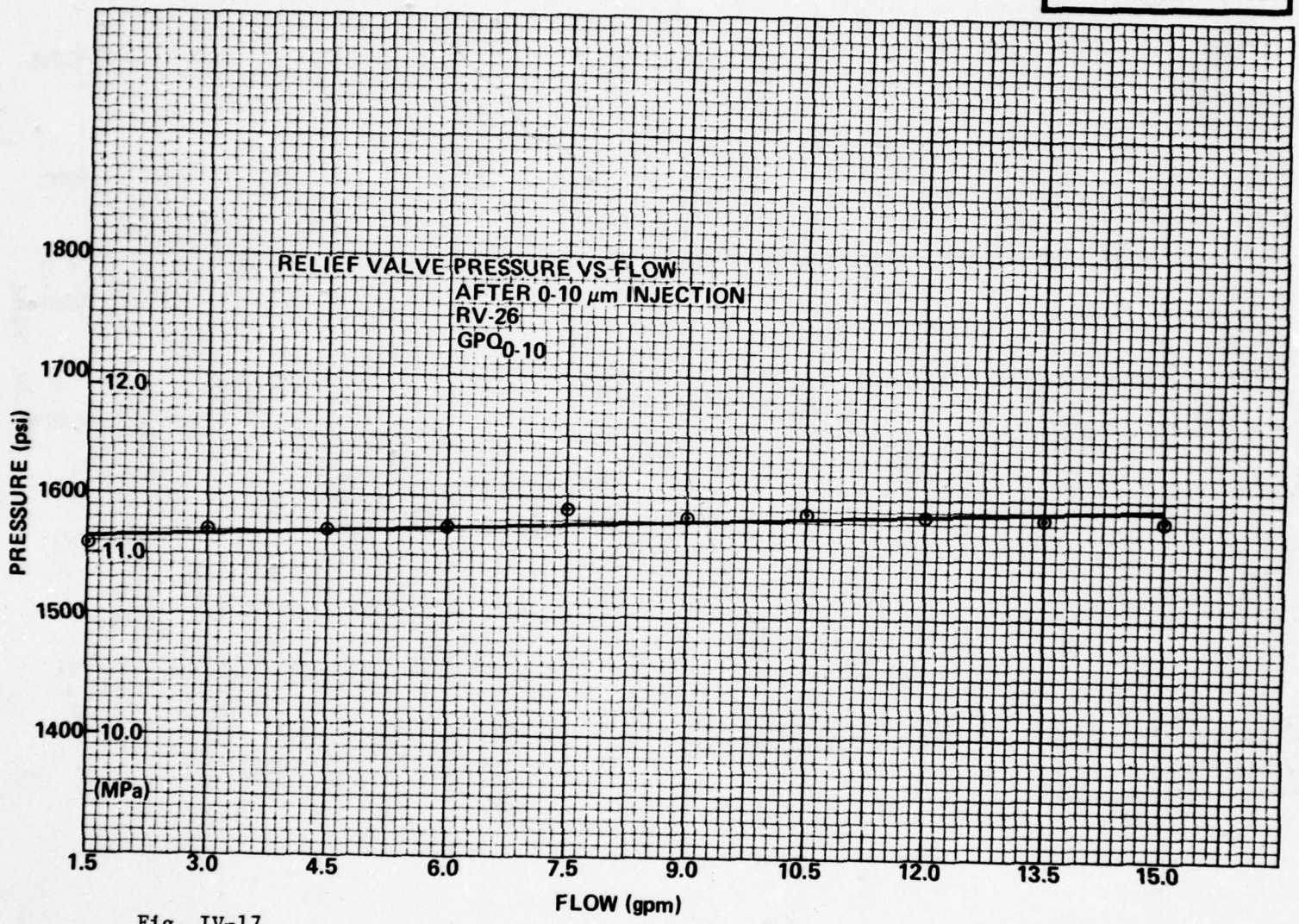


Fig. IV-17

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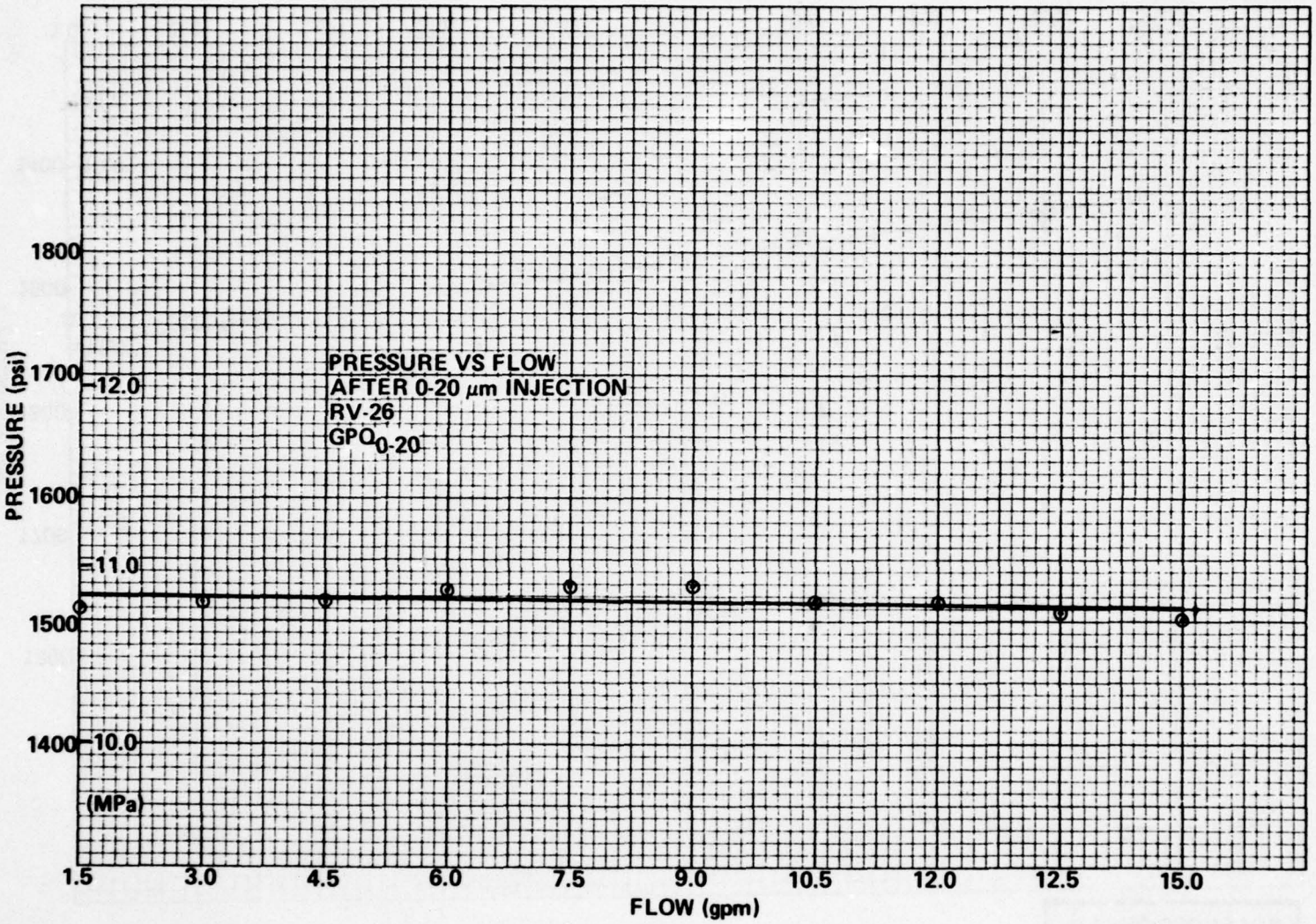


Fig. IV-18

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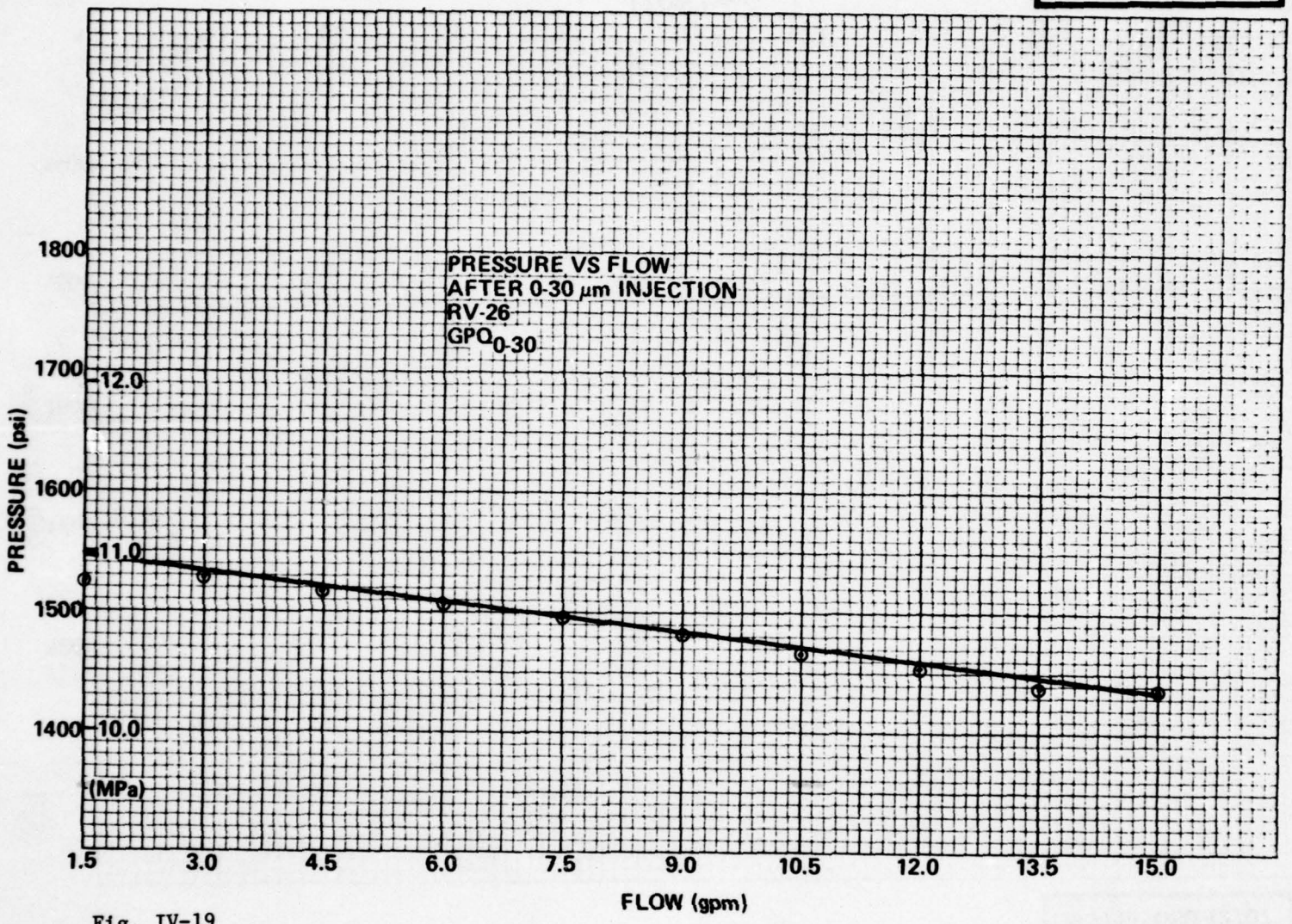


Fig. IV-19

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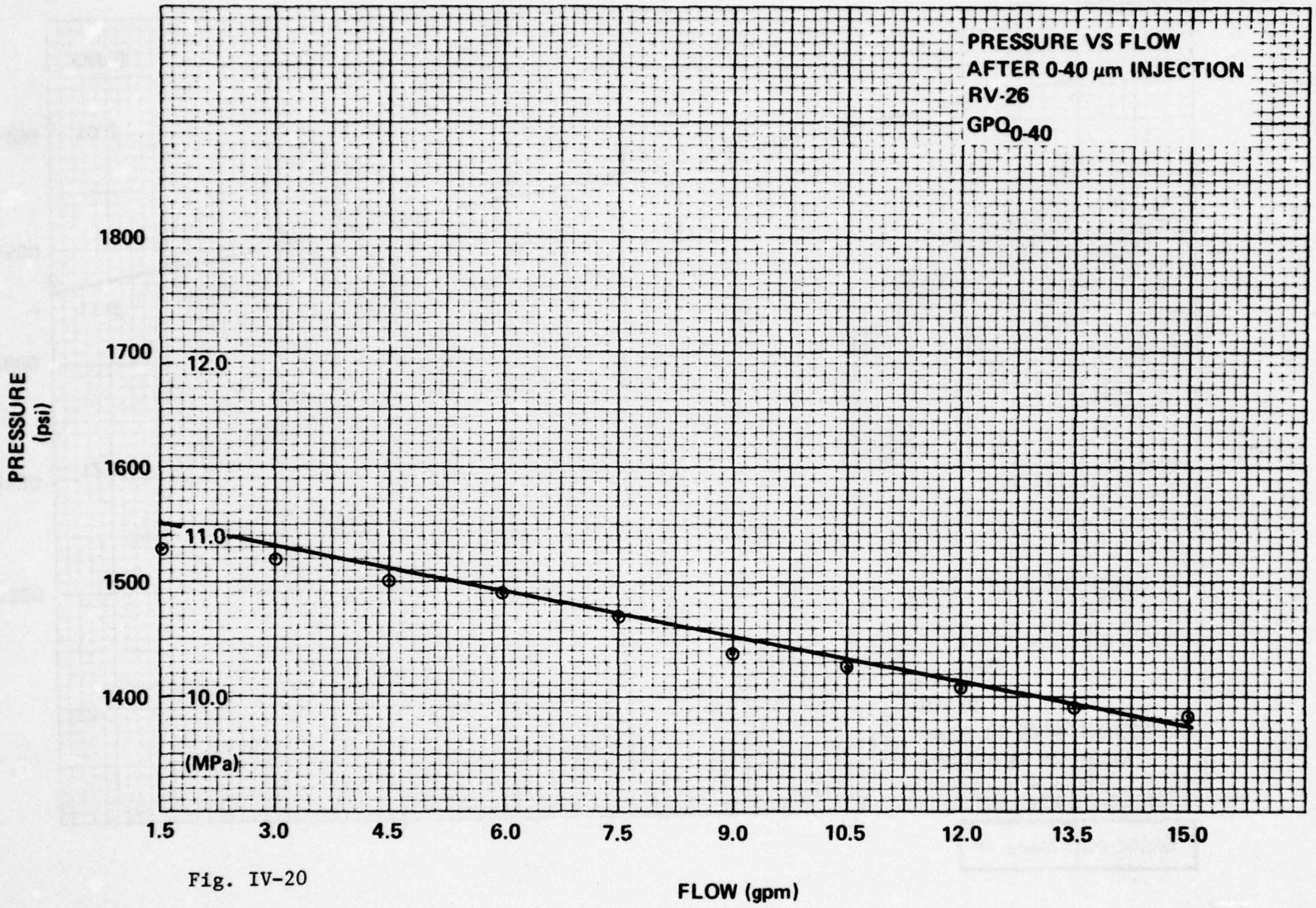


Fig. IV-20

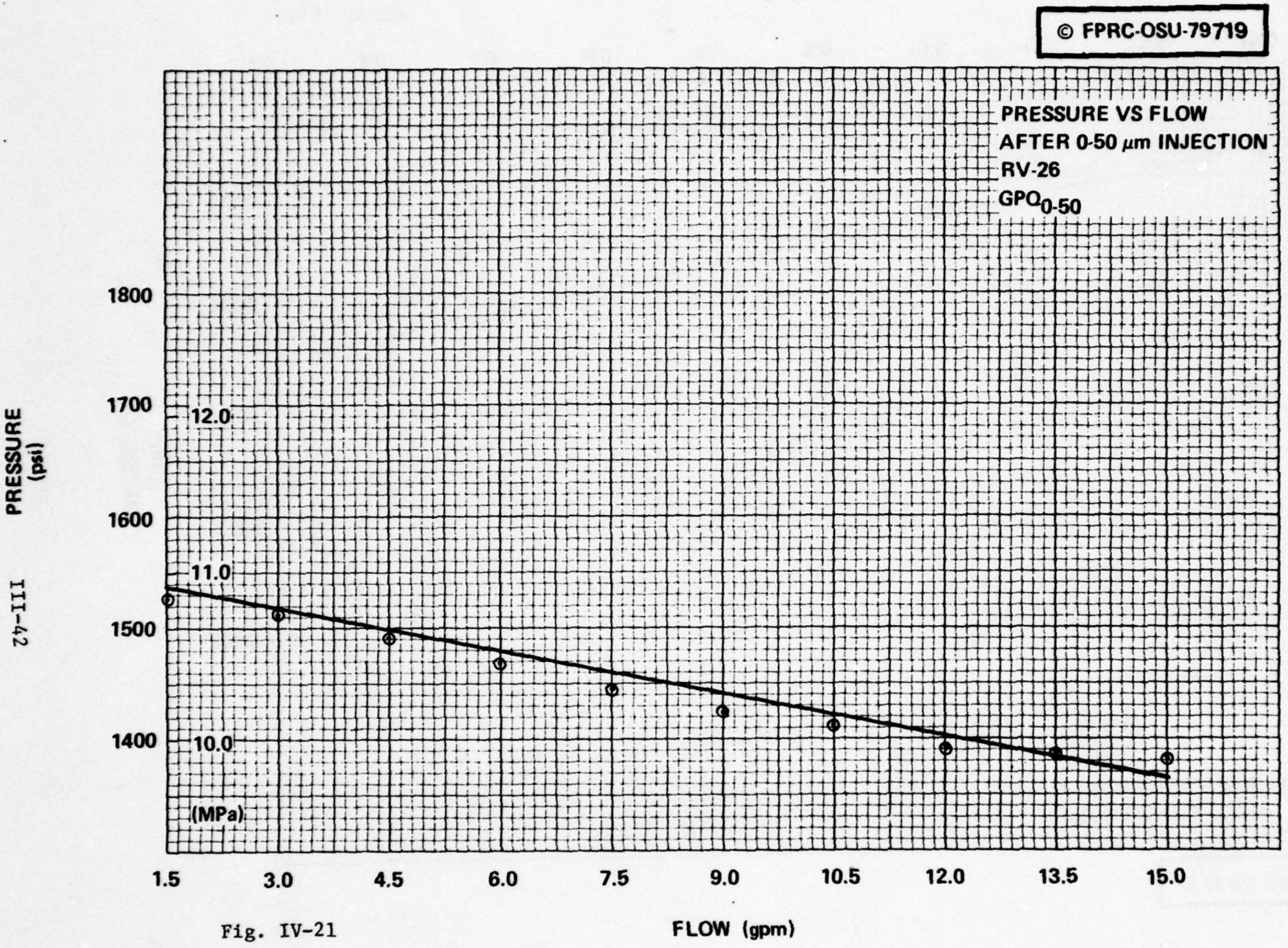

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PRESSURE VS FLOW
AFTER 0-50 μm INJECTION
RV-26
GPO0-50
PRESSURE (psi)
1800
1700
1600
1500
1400
12.0
11.0
10.0
(MPa)
1.5
3.0
4.5
6.0
7.5
9.0
10.5
12.0
13.5
15.0
FLOW (gpm)

Fig. IV-21

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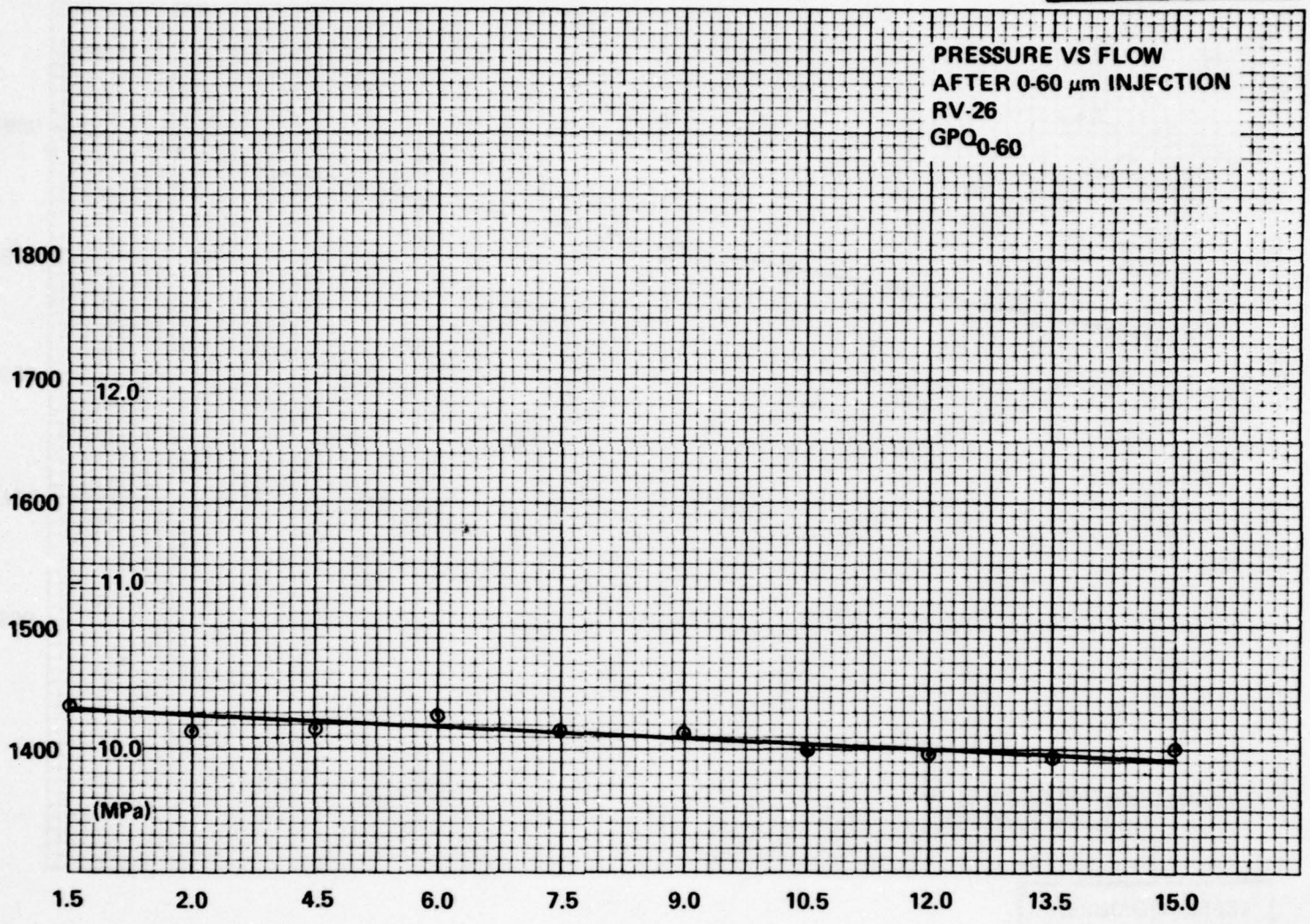


Fig. IV-22

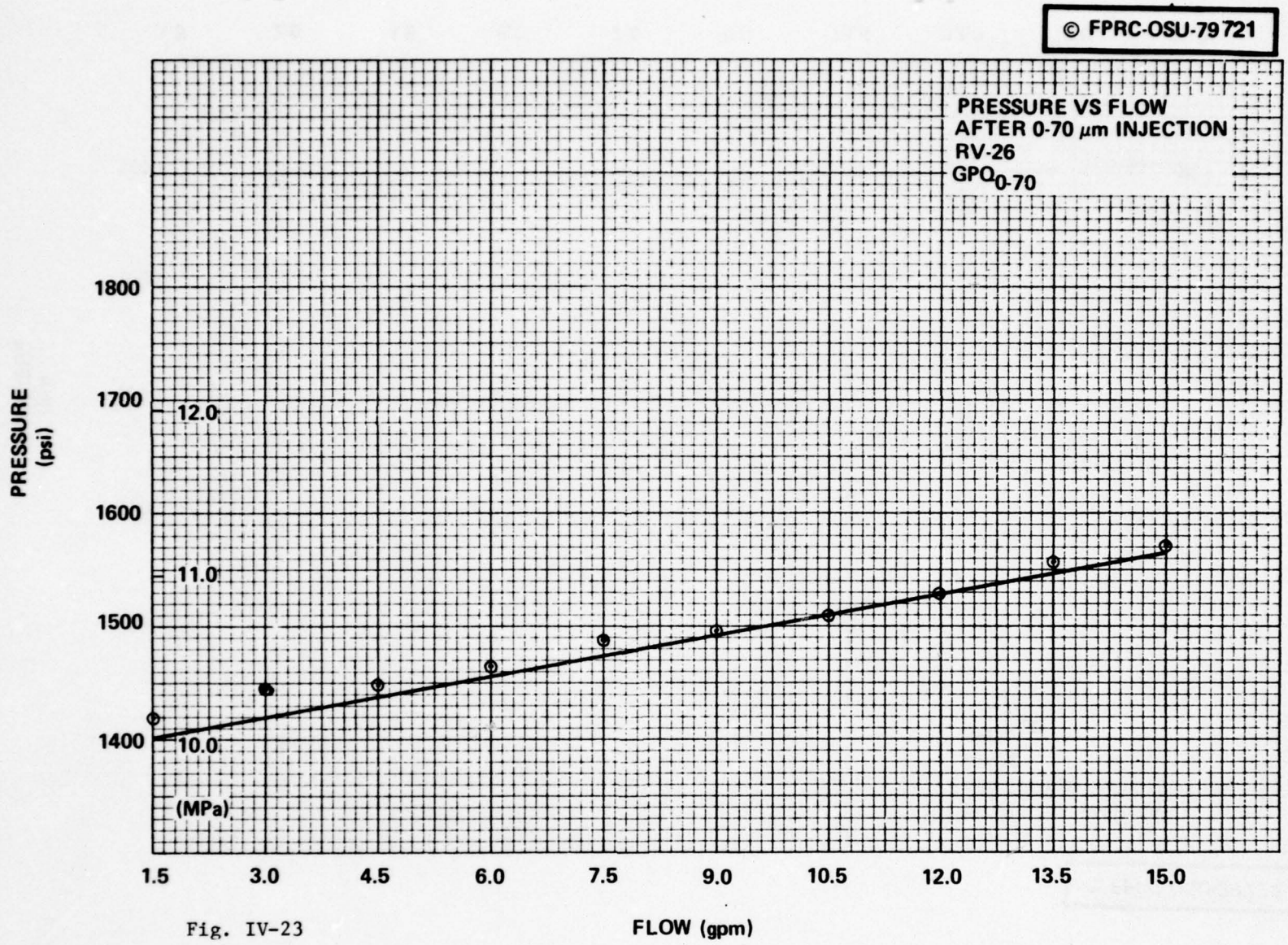

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PRESSURE VS FLOW
AFTER 0-70 μm INJECTION
RV-26
GPQ0-70
PRESSURE (psi)
1800
1700
1600
1500
1400
12.0
11.0
10.0
(MPa)
1.5
3.0
4.5
6.0
7.5
9.0
10.5
12.0
13.5
15.0
FLOW (gpm)

Fig. IV-23

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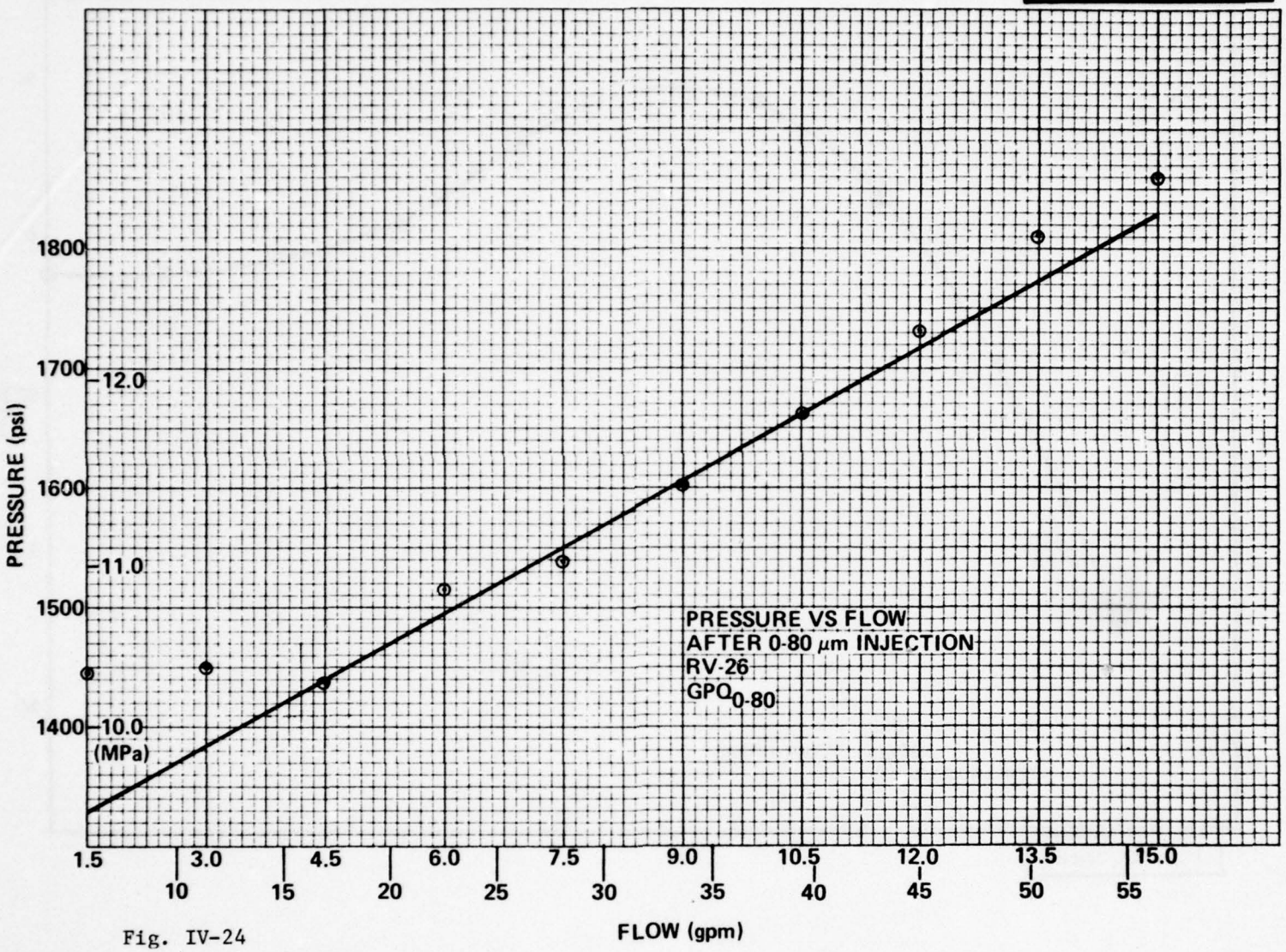


Fig. IV-24

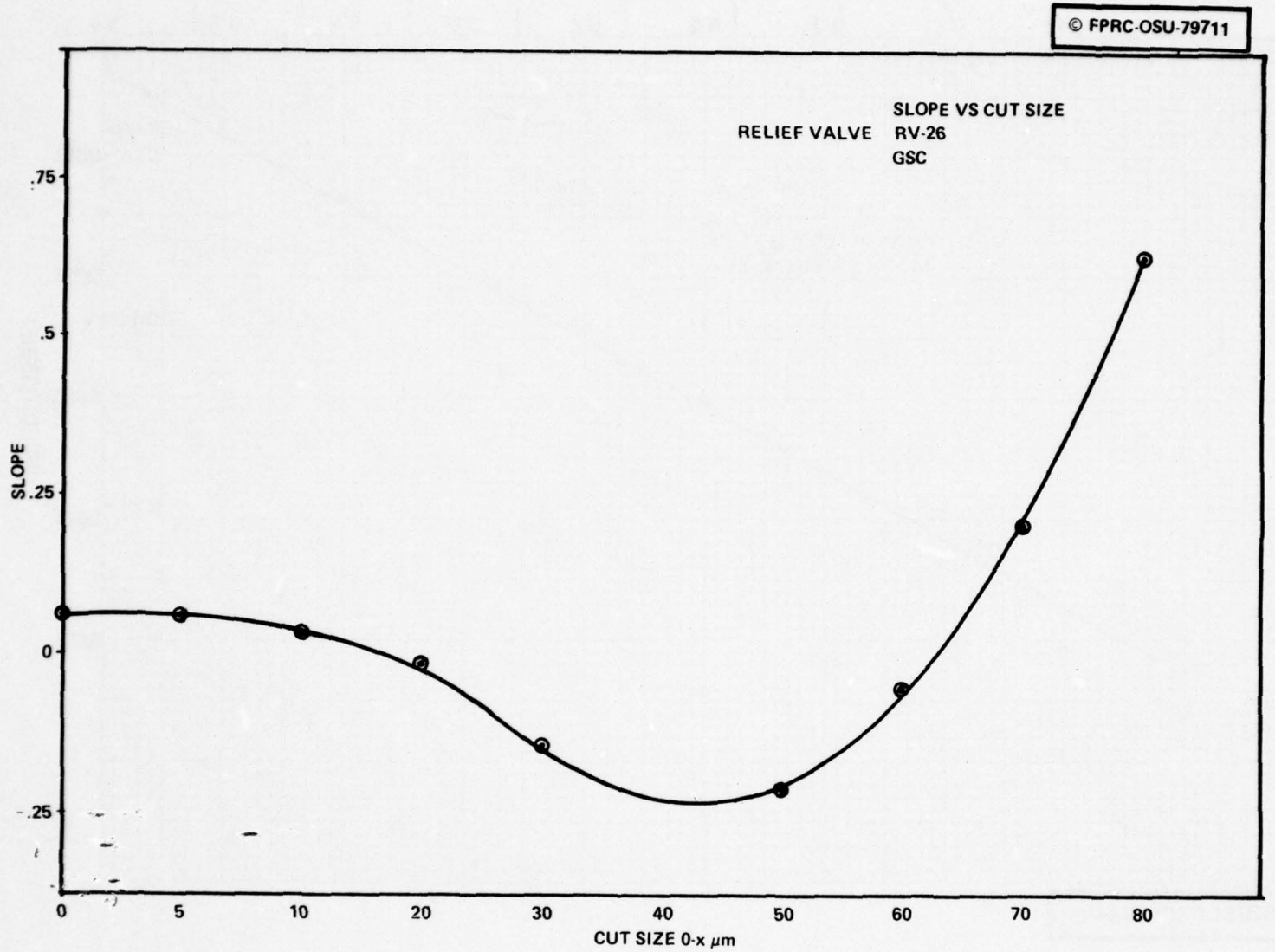

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SLOPE VS CUT SIZE
RELIEF VALVE RV-26
GSC
.75
.5
.25
0
-.25
SLOPE
0
5
10
20
30
40
50
60
70
80
CUT SIZE 0-x μm

Fig. IV-25

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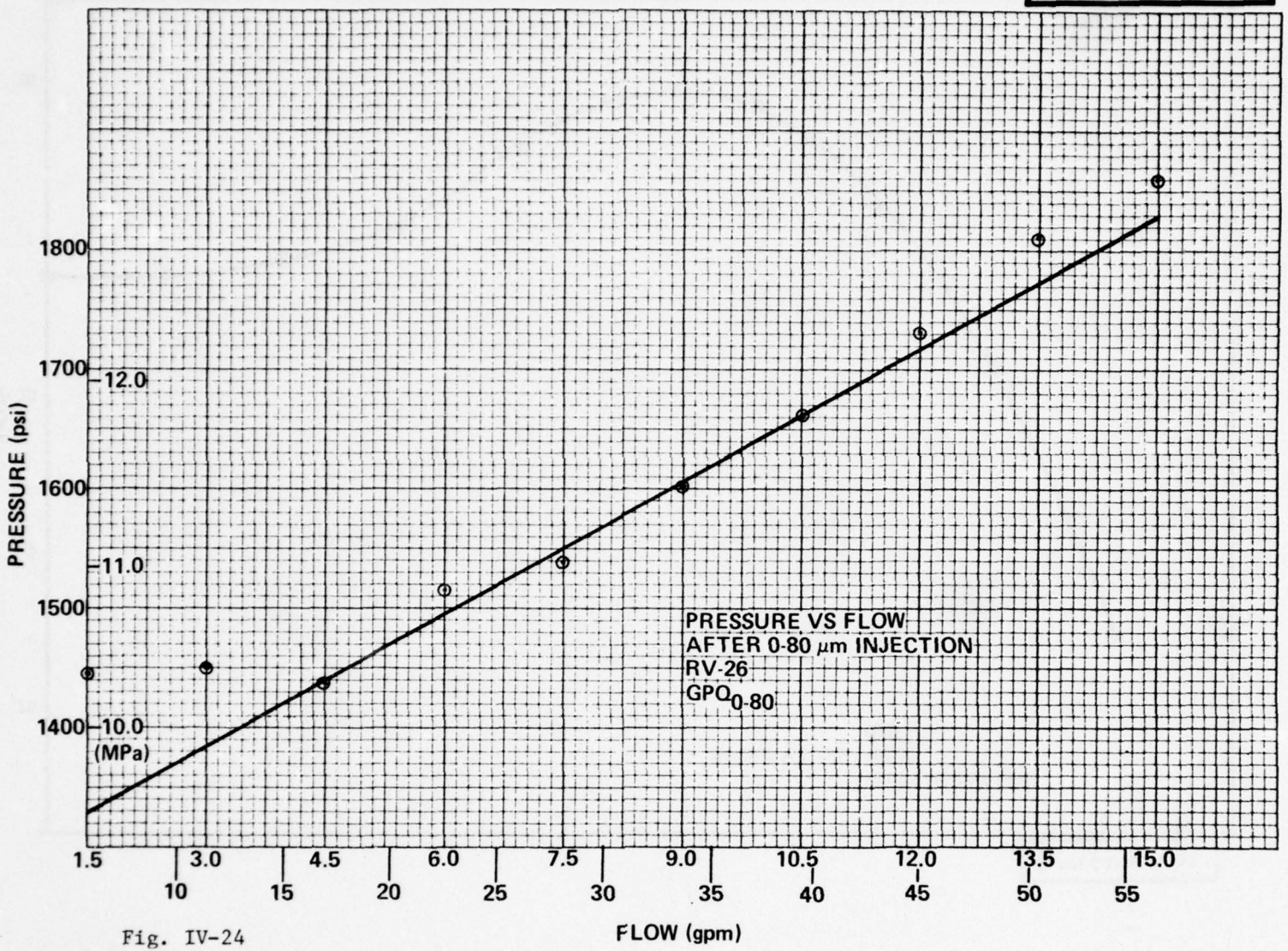


Fig. IV-24

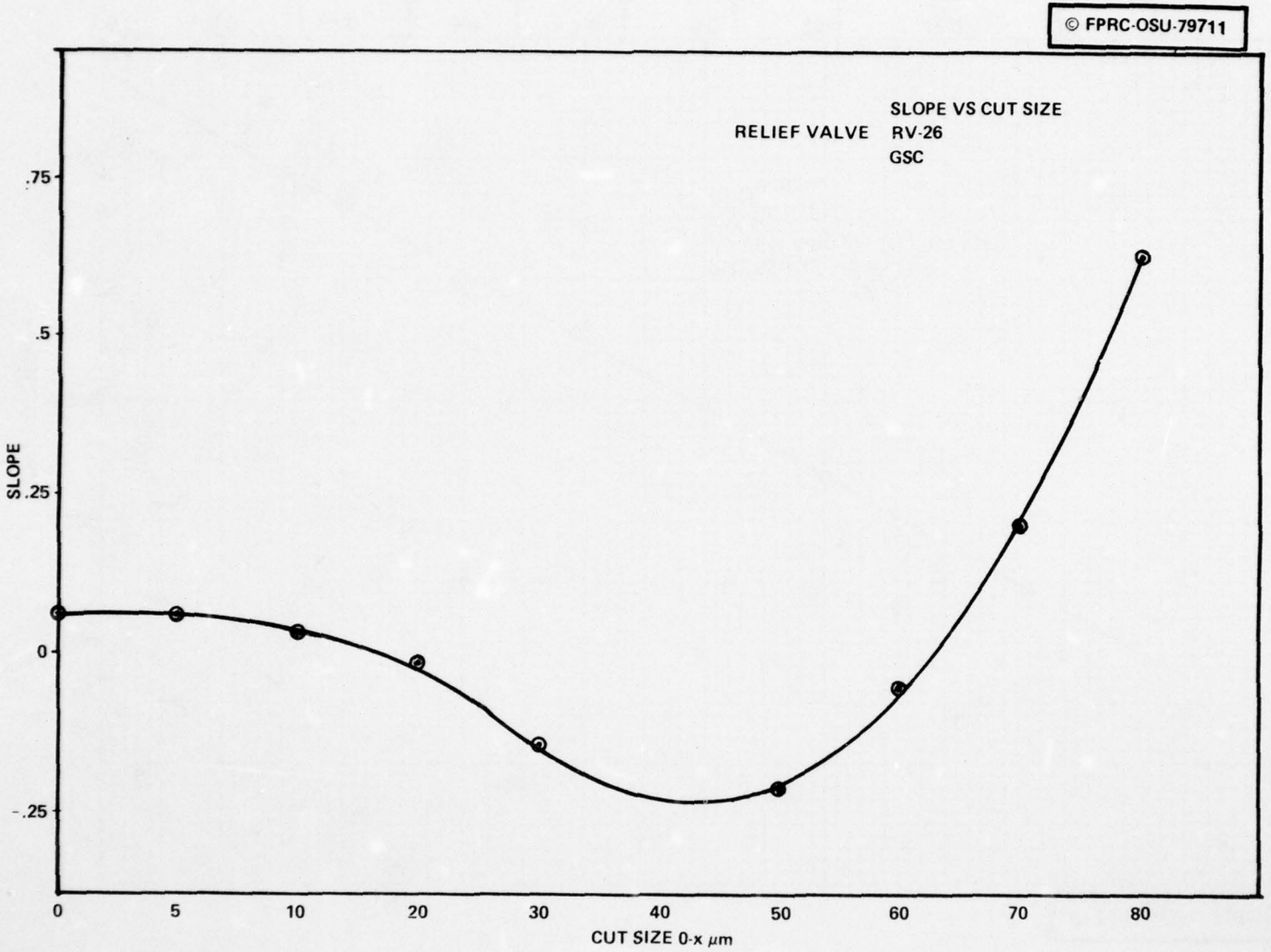

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SLOPE VS CUT SIZE
RELIEF VALVE RV-26
GSC
.75
.5
.25
0
-.25
SLOPE
0
5
10
20
30
40
50
60
70
80
CUT SIZE 0-x μm

Fig. IV-25

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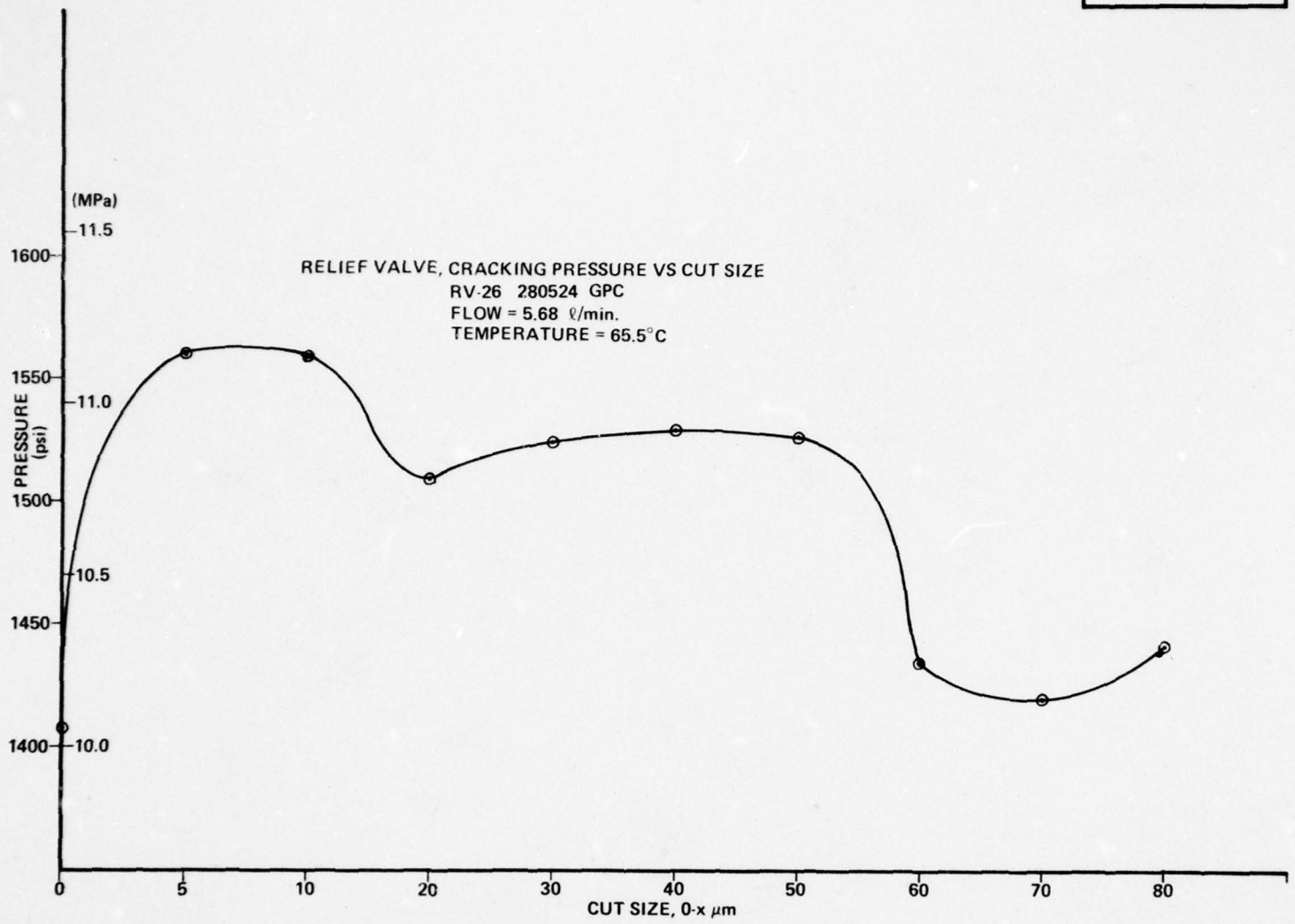


Fig. IV-26

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

Relief valves are critical components of most fluid power systems and need to be suitably selected and protected from particulate contaminants for satisfactory and reliable operation. The development and verification of a realistic, repeatable and reproducible test procedure is the first step in improving component selection, maintenance practices, diagnostics and ultimate diagnostics. This report has discussed the rationale for and established the practicality of the test procedure for pilot operated relief valves. The test procedure follows the general philosophy of pump and motor contaminant sensitivity testing, which has gained industrial acceptance. In fact, the same test stand, with minor changes can be used for pump motor or valve tests. The performance parameters which are used as a measure of degradation are the cracking pressure and the average slope of the pressure-flow curve. Changes in both need to be recorded since they are independent, and different applications may emphasize one or the other parameter.

Though the test procedure was drafted to pilot operated relief valves, it can be used, with minor editorial modifications, for single stage relief valves. Relief valves fall in the broad category of two port valves, and with some changes in the definition and measurement of the performance appraisal parameters, the same procedure can be used for the whole category. Thus, needle valves, ball valves and check valves can be examined for contaminant sensitivity with the same test circuit. The testing of valves with three ports, eg. flow control valves and flow dividers, needs changes

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in not only the test circuit but also the method of obtaining the static characteristics. Once again it may not be realistic to use a single parameter to describe changes in performance due to contaminant exposure.

Valves with four or more ports are generally directional control valves, and test procedures for appraising their contaminant sensitivity will be somewhat different from that for pressure and flow control valves. Directional control valves can be classified by number of spool positions, port connections in the neutral positions, actuating method, number of detents, centering method, and type of moving element (spool, plug, poppet, etc.). The performance parameter and the test set-up could conceivably be different for each category. Additional complications arise when multiple spools are enclosed in one housing, and flow paths may involve more than one spool. Such valves usually have complex flow galleries which can trap contaminants, thus rendering control of the contamination level rather difficult. In summary, it is recommended that a test procedure be drafted for each category of valves. The scope and purpose of each procedure should be clearly indicated in each document so as to ensure that is is not mis-applied.

The process of contaminant degradation of hydraulic components is stochastic in nature, ie. there is no way of assuring that two successive tests on identical components will result in exactly the same changes in performance parameters. Hence it is imperative, especially during design qualifications tests, to test enough sample to establish the changes in a statistical sense.

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1. Iyengar, S.K.R. and R.F. Sharp, "The Effect of Particulate Contaminants on the Static Characteristics of Fluid Power Valves," ASME Paper 77-DE-20, presented at the Design Engineering Conference and Show, Chicago, IL, May 9-12, 1977.

2. Iyengar, S.K.R., "Static and Dynamic Performance Degradation of Compound Relief Valves," The BFPR Journal 1978, 11, pp 159-164.

3. Annual Report, MERADCOM/OSU Hydraulic System Reliability Program, Section III, FPRC-7M3 prepared by Fluid Power Research Center, Oklahoma State University, Stillwater, OK, 1977.

4. Annual Report, Hydraulic Specification Program (V), Phase II, AD 737 730, 1971.

5. Recommended Procedures for Evaluating fluid Power Components and Systems, FPRC 72-1, Fluid Power Research Center, Oklahoma State University, Stillwater, OK, 1972

6. Maroney, G. E. and E. C. Fitch, "A Fundamental Method for Evaluating the Contaminant Tolerance of Fluid Power Control Valves," Proc. NCFP, Vol. XXVII, 1972, pp 548-562.

APPENDIX A

PROCEDURE FOR EVALUATING THE CONTAMINANT SENSITIVITY OF A FLUID POWER PILOT OPERATED PRESSURE RELIEF VALVE

Introduction

In hydraulic fluid power systems, a pressure relief valve is required to operate satisfactory in a contaminated environment. The contaminant sensitivity of a pressure relief valve is a mandatory consideration in determining the acceptable contamination level in which the valve performs satisfactorily.

1. Scope

   To include the evaluation of the contaminant sensitivity of a hydraulic pilot operated pressure relief valve when it is subjected to a given size distribution of test contaminant. To monitor degradation of a performance parameter at specific operating conditions as the evaluation criterion. Valve failure by processes other than the performance parameter are not considered in this procedure.

2. Purpose

   To ascertain a uniform, consistent and economical method of evaluating the contaminant sensitivity characteristics of a hydraulic pilot operated pressure relief valve.

3. Terms (For definitions of terms not herein defined, see Reference No. 1).

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3.1 Specified Flow. Steady-State flow rate for the component as specified by the manufacturer.

3.2 Specified Pressure. Stead state operating pressure for the component as specified by the manufacturer.

3.3 Contaminant Sensitivity. The degradation in performance of a hydraulic component with respect to the fluid contamination level.

3.4 Gravimetric Level. The quantity of contaminant by weight per unit volume of fluid.

3.5 Cracking Pressure. Pressure at which the valve passes one-twentieth of the specified flow of the valve.

4. Units

4.1 The international system of units (SI) is used in accordance with Reference No. 2.

4.2 Approximate conversions to "US INCH" units are given for information purposes; these appear in parenthesis after their SI counterpart.

5. Graphic Symbols

Graphic symbols used herein are in accordance with Reference Nos. 3 and 4. Where Reference No. 4 is not in agreement with No. 3, Reference No. 3 shall govern.

6. Letter Symbols

bar or pascal	pressure
cSt	centistokes (viscosity)
^{o}C	degrees Celsius (temperature)
CSI	Contaminant Sensitivity Index
G_i	quantity of test contaminant of size range 0-1 micrometres, required to establish a 300 mg/l gravimetric level in the test system (gram)
l/min	litres per minute
Σ	Summation

7. Equipment

7.1 Hydraulic Test Circuit. The test system volume shall be numerically equal to ½ of the specified flow of the valve per minute (See Fig. 1).

7.2 Automatic particle counter calibrated per NFPA Recommended Standard T2.9.6.

7.3 Supply of classified, AC Fine Test Dust Contaminant.

7.4 Supply of super-clean fluid sample bottles.

7.5 Supply of clean slurry injection bottles.

7.6 The control filter shall be capable of providing a contaminant background of less than 100 particles per millilitre greater than 10 micrometres.

7.7 The test facility shall be capable of maintaining a distribution as verified in the following manner: Inject an amount

of contaminant calculated to establish a particle distribution equivalent of 5 mg/litre. During a 30 minute period of operation of the test facility, at ½ of the specified flow of the valve to be tested, obtain four fluid samples, 2 minutes apart. At any particle size (5μM-80μM), no particle count shall vary from the average of the individual particle counts by more than 10 percent.

8. Test Conditions

8.1 The test temperature shall be specified by the test requirement data.

8.2 The test fluid shall be specified by the test requirement data.

8.3 Data Accuracy is given in Table A-1.

TABLE A-1.

Quantity	Customary Units	SI Units	Maintain within (±) of Actual
Pressure, Inlet	PSI	bar	2%
Pressure, Outlet	PSI	bar	2%
Flow	USGPM	l/min	2%
Temperature	^{o}F	^{o}C	$5^{o}F$ ($3^{o}C$)
Volume	Gallons	Litres	2°

9.0 Test Procedure

9.1 Preliminary Preparation

9.1.1 The valve shall be mounted on the test apparatus as shown in Fig. A-1.

9.1.2 Circulate the system fluid through the valve and control filter until a background level of not more than 100 particle per millilitre greater than 10 μM is attained and the system temperature has stabilized at the prescribed valve.

9.1.3 Adjust the system flow rate to be one-half that of the valves specified flow.

9.1.4 Adjust the valve to the desired pressure setting for the test. Ensure that the adjusting mechanism is locked, so as not to be distrubed by vibration.

9.1.5 Record the differential pressure across the valve and flow through the valve.

9.2 Performance Parameter Measurement

9.2.1 Adjust the flow through the valve to one-twentieth the specified flow.

9.2.2 Slowly increase the flow through the valve.

9.2.3 Record the differential pressure across the valve at ten equal flow points from one-twentieth flow to one-half specified flows. Flow rate shall at no time be reduced between the measurement points.

9.2.4 Record this as PQ_I.

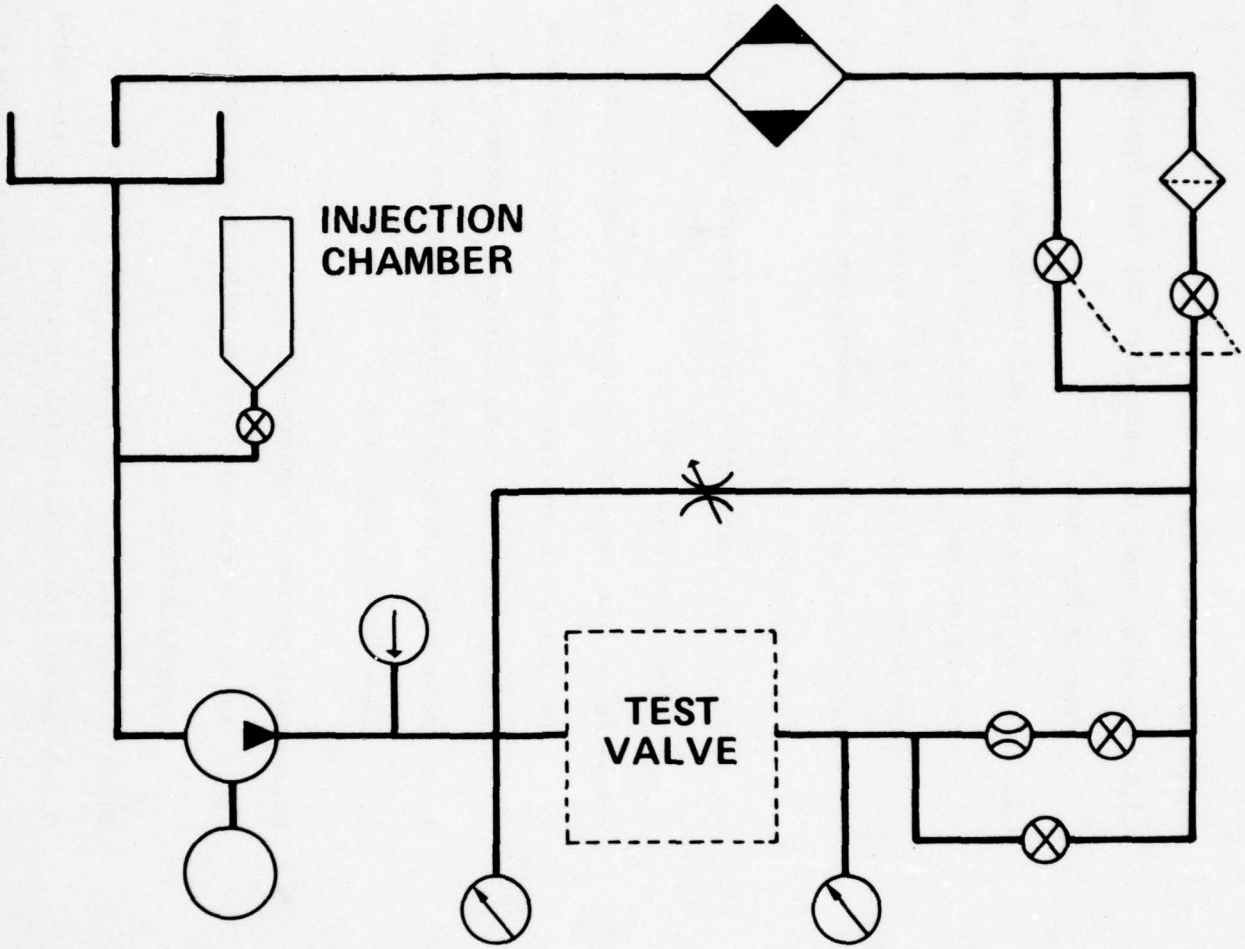


Fig. A-1

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9.2.5 Break-in the valve as follows:

9.2.5.1 Operate the valve at half specified flow for ten minutes, then slowly decrease the flow below one-twentieth specified flows.

9.2.5.2 Perform section 9.2.3.

9.2.5.3 Record this as PQ_{II}.

9.2.5.4 Perform section 9.2.5.1.

9.2.5.5 Perform section 9.2.3.

9.2.5.6 Record this as PQ_S.

9.3 Contaminant Sensitivity Test

9.3.1 Bypass the control filter.

9.3.2 Prepare an oil slurry of each of the following contaminant ranges: (each containing G_i grams) 0-5μM, 0-10μM, 0-20μM, 0-30μM, 0-40μM, 0-50μM, 0-60μM, 0-70μM, and 0-80μM.

9.3.3 Inject the oil slurry and distribute it in the system at low pressure for 1 minute. Circulate the fluid through the system at specified pressure and one-half specified flow rate for thirty minutes. If the valve cannot maintain the pressure between 50% and 150% of the original pressure setting, terminate the test.

9.3.4 Circulate the fluid through the control filter until a background level of less than 100 particles per millilitre greater than 10μM or less is attained.

9.3.5 Repeat section 9.2.1 through 9.2.3.

9.3.6 Record this measurement as PG_{0-X} where X is the maximum size of contaminant in the injection.

9.3.7 Bypass the control filter.

9.3.8 Prepare the next oil slurry.

9.3.9 Repeat sections 9.3.4 through 9.3.9 until the 0-80μM oil slurry has been injected, or the cracking pressure has changed by 50%.

9.4 Reporting Method

9.4.1 Record the valve I.D. number, specified flow rate, and specified pressure.

9.4.2 On standard graph paper, plot the pressure versus flow for the data recorded in section 9.2.4. Label this graph as GPQ_I.

9.4.3 Perform Section 9.4.2 for the data recorded in section 9.2.8. Label this graph as GPQ_{II}.

9.4.4 Perform Section 9.4.2 for the data recorded in section 9.2.10. Label this graph as GPQ_S.

9.4.5 Perform Section 9.4.2 for each set of data recorded in section 9.3.6. Label these graphas as GPQ_{o-x}, where x is the maximum size of contaminant in the injection.

9.4.6 For each graph of data in section 9.4.4 and section 9.4.5, best fit a straight line through the points. Record the slope of each line.

9.4.7 On standard graph paper, plot the one-twentieth flow pressure point from the data recorded in section 9.2.5.6

and section 9.3.7 versus the maximum size of contaminant in the injection. Label this graph as GPC.

9.4.8 On standard graph paper, plot the slopes of the lines in section 9.4.6 versus the maximum size of contaminant in the injection. Label this graph as GSC.

REFERENCES

1. American National Standard Glossary of Terms for Fluid Power ANSI/B93.2-1971, and Supplements thereto, (ISO/TC 131/SC 1) (USA-2 3).
2. SI units and recommendations for the use of their multiples and of certain other units, ISO 1000-1973.
3. American National Standard Drafting Practices for Fluid Power Diagrams, ANSI/Y14.17-1966.
4. International Standard Graphical Symbols for Hydraulic and Pneumatic Equipment and Accessories for Fluid Power Transmission, ISO/R 1219-1970. Agrees with ANSI/Y32,10-1967.

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APPENDIX B

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QUESTIONNAIRE ON RELIEF VALVES

This questionnaire should be considered only with respect to Relief Valves (hereafter called valves) and their contaminant sensitivity.

1. Should "cracking" pressure be specified as the pressure at which some percentage of flow is passed?

   Yes _____

   No _____ if not, define "cracking" pressure _____

   _____

   If yes, what percentage of rated flow?

   1% _____ 2% _____ 5% _____ 10% _____ Other _____

   Comments _____

   _____

2. Is leakage, below cracking pressure, a valid test parameter?

   Yes _____

   No _____

   Comments _____

   _____

3. Is some leakage, below cracking pressure, acceptable?

   Yes _____

   No _____

   If yes, what percentage of rated flow is acceptable?

   1/4% _____ 1/2% _____ 1% _____ 2% _____ Other _____

   Comments _____

   _____

RELIEF VALVES

4. Is the change in pressure drop, from cracking to rated flow, a valid test parameter?

   Yes _____

   No _____

   Comments _____

   _____

5. Is pressure drop vs flow a valid test parameter?

   Yes _____

   No _____

   Comments _____

   _____

6. Do you feel that cracking and reseating pressures are valid performance parameters?

   Yes _____

   No _____

   Why or why not _____

   _____

7. Is maximum static relief valve pressure a good performance parameter?

   Yes _____

   No _____

   Comments _____

   _____

8. Is peak overshoot pressure a good performance parameter for contaminant sensitivity tests?

   Yes _____

   No _____ if not, why not _____

   If yes, at what pressure rise rate?

   30,000 psi/sec _____ 60,000 psi/sec _____

   100,000 psi/sec _____ Other _____

   Comments _____

   _____

9. Is time to steady state pressure a valid contaminant test parameter? (After having a certain pressure rise rate)

   Yes _____

   No _____

   Comments _____

   _____

10. Should hysteresis be considered a valid parameter for relief valve tests?

    Yes _____

    No _____

    Comments _____

    _____

11. During a valve contaminant sensitivity test parameters will vary. In order to insure reasonable accuracy how closely should the following be controlled?

RELIEF VALVES

Acceptable Variation	0-2%	3-5%	6-10%	More
Pressure at inlet	____	____	_____	____
Pressure at outlet	____	____	_____	____
Pressure at working ports	____	____	_____	____

Other _____

Comments _____

_____

12. What type of contaminant(s) should be used in valve contaminant sensitivity tests?

ACFTD _____

Alluminum _____

Carbonyl/Iron _____

Other _____

Comments _____

_____

13. Is the above contaminant(s) checked representative of what would be encountered by the valve in normal use?

Yes _____

No _____ if not, why _____

14. What do you feel the Background Contamination Level should be for evaluation of the operating characteristics of a valve after exposure to contaminant. (Performance testing is done with "clean" oil. How clean is "clean"?)

60 particles/ml>10 μm _____

100 particles/ml>10 μm _____

500 particles/ml>10 μm _____

1000 particles/ml>10 μm _____

2000 particles/ml>10 μm _____

Other _____

Comments _____

_____

15. Should a valve contaminant sensitivity test expose the valve to a fine contaminant at the beginning of the test with maximum particle sizes increasing stepwise as the test is performed?

Yes _____

No _____

Comments _____

_____

16. What should be the minimum contaminant cut size for a valve test?

0-5 μm _____

0-10 μm _____

0-20 μm _____

0-30 μm _____

Other _____

Comments _____

_____

RELIEF VALVES

17. What do you feel should be the maximum contaminant cut size for a valve test?

0-20 µm _____

0-30 µm _____

0-40 µm _____

0-60 µm _____

0-80 µm _____

Other _____

Comments _____

_____

18. What do you feel should be the minimum gravimetric level for valve tests? 300 mg/l is standard for Motor and Pump tests

10 mg/l _____

20 mg/l _____

40 mg/l _____

60 mg/l _____

100 mg/l _____

300 mg/l _____

Other _____

Comments _____

_____

19. What do you feel should be the maximum gravimetric level for valve tests?

20 mg/l _____

40 mg/l _____

60 mg/l _____

100 mg/l _____

300 mg/l _____

600 mg/l _____

Other _____

Comments _____

_____

20. Accelerated contaminant sensitivity tests involve higher levels of contaminant than found in most systems. Do you feel that accelerated contaminant tests are a valid measure of contaminant sensitivity?

Yes _____

No _____

Why _____

21. Should there be various durability ratings of relief valves corresponding to how they may be used? (safety only, intermitent, continuous)

Yes _____

No _____

Comments _____

_____

22, Should relief valves be tested by exposing them to pressures above normal at a fixed duration at a fixed frequency?

Yes _____

No _____

Comments _____

_____

RELIEF VALVES

23. What pressure rise rate should be used for cyclic testing? (may be different from #8)

    10,000 psi/sec _____

    20,000 psi/sec _____

    30,000 psi/sec _____

    60,000 psi/sec _____

    Other _____

    Comments _____

    _____

24. Should the valve be made to pass half it's rated flow during cycling, with pressure riase rate limited as in #22?

    Yes _____

    No _____

    Comments _____

    _____

25. Would one second cycles be acceptable? (1/2 sec flow, 1/2 sec nonflowing)

    Yes _____

    No _____

26. Should standard contaminant sensitivity tests be available for testing under nonstandard conditions? (Heat, cold, fire resistant fluid)

    Yes _____

    No _____

QUESTIONNAIRE ON MANUALLY OPERATED, 4 WAY, SPOOL DIRECTIONAL CONTROL VALVES

This questionnaire should be considered only with respect to Manually Operated, 4 Way, Spool Directional Control Valves (hereafter referred to as valves) and their contaminant sensitivity.

1. During a valve contaminant sensitivity test parameters will vary. In order to insure reasonable accuracy how closely should the following be controlled?

Acceptable Variation	0-2%	3-5%	6-10%	More
Pressure at inlet	____	____	_____	____
Pressure at outlet	____	____	_____	____
Pressure at working ports	____	____	_____	____

Other _____

Comments _____

_____

2. What type of contaminant(s) should be used in valve contaminant sensitivity tests?

ACFTD _____

Alluminun _____

Carbonyl/Iron _____

Other _____

Comments _____

_____

3. Is the above contaminant(s) checked representative of what would be encountered by the valve in normal use?

Yes _____ No _____ If not, why _____

4. What do you feel the Background Contaminantion Level should be for evaluation of the operating characteristics of a valve after exposure to contaminant. (Performance testing is done with "clean" oil. How clean is "clean"?)

   60 particles/ml>10 µm _____

   100 particles/ml>10 µm _____

   500 particles/ml>10 µm _____

   1000 particles/ml>10 µm _____

   2000 particles/ml>10 µm _____

   Other _____

   Comments _____

   _____

5. Should a valve contaminant sensitivity test expose the valve to a fine contaminant at the beginning of the test with maximum particle sizes increasing stepwise as the test is performed?

   Yes _____

   No _____

   Comments _____

   _____

6. What should be the minimum contaminant cut size for a valve test?

   0-5 µm _____

   0-10 µm _____

   0-20 m _____

   0-30 m _____

   Other _____

   Comments _____

MANUALLY OPERATED, 4 WAY, SPOOL DIRECTIONAL CONTROL VALVES

7. What do you feel should be the maximum contaminant cut size for a valve test?

   0-20 µm _____

   0-30 µm _____

   0-40 µm _____

   0-60 µm _____

   0-80 µm _____

   Other _____

   Comments _____

   _____

8. What do you feel should be the minimum gravimetric level for valve tests? 300 mg/l is standard for Motor and Pump tests

   10 mg/l _____

   20 mg/l _____

   40 mg/l _____

   60 mg/l _____

   100 mg/l _____

   300 mg/l _____

   Other _____

   Comments _____

   _____

9. What do you feel should be the maximum gravimetric level for valve tests?

   20 mg/l _____

   40 mg/l _____

60 mg/l _____

100 mg/l _____

300 mg/l _____

600 mg/l _____

Other _____

Comments _____

_____

10. Accelerated contaminant sensitivity tests involve higher levels of contaminant than found in most systems. Do you feel that accelerated contaminant tests are a valid measure of contaminant sensitivity?

Yes _____

No _____

Why _____

11. Do you feel that changes in operating force should be monitored during a contaminant sensitivity test?

Yes _____

No _____ if not, why _____

Comments _____

_____

12. What parameters reguarding operating force should be measured?

Maximum operating force _____

Operating force vs displacement _____

Other _____

Comments _____

_____

13. Is Mil 2104 oil acceptable for valve testing?

Yes _____

No _____ if no, why _____

What oil should be used? _____

14. At what temperateure should the oil be during standard valve tests?

Less than 120° F _____

120° - 150° F _____

150° - 180° F _____

180° - 210° F _____

Greater than 210°F _____

Comments _____

_____

15. Is silting a valve performance parameter?

Yes _____

No _____ if not, why _____

16. If silting is a valve performance parameter, how long should a silting test be run before evaluation? _____

17. Is leakage a valid performance parameter?

	Yes	No
External	___	___
Internal	___	___

Comments _____

_____

MANUALLY OPERATED, 4 WAY, SPOOL DIRECTIONAL CONTROL VALVES

18. In what spool positions should leakage be tested?

Full in _____

Centered _____

Full out _____

Comments _____

_____

19. Is percentage of rated flow a valid base for comparison of internal leakage?

Yes _____

No _____ if not, why _____

20. If yes, what percentage of rated flow is acceptable?

<1/2%___ 1/2%___ 1%___ 2%___ 5%___ >5%___

Comments _____

_____

21. Should there be different classes of valves, dependent upon expected frequency of use. (Ex: on "intermitent" rating text X hrs and a "continuous" rating for those tested WY hrs.)

Yes _____

No _____

Comments _____

_____

22. Should standard contaminant testing procedures be available for valve evaluation under unusual conditions? (heat, cold, various environments, fire resistant fluids)

Yes _____

No _____

Comments _____

_____

SECTION IV

COMPONENT DEGRADATION ASSESSMENT

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PREFACE

This section presents a detailed account of the project activities in the area of Component Degradation Assessment. The objective of this activity is to determine a methology for determining component residual life. Both theoretical and laboratory investigations were performed.

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CHAPTER I

COMPONENT DEGRADATION ASSESSMENT

The ability for a mobile machine to complete its intended mission is of primary importance in selection and scheduling of machines to perform a given task. For example, a machine whose performance has recently degraded until only minimum performance specifications are met is an unlikely candidate for an extended severe mission. Because of the often sudden degradation in performance, the determination of mission worthiness is usually not obvious.

An accurate knowledge of component residual life is a key factor in evaluating mission worthiness of a mobile machine. Certainly the residual life of a machine should be considered prior to assigning it to a given task. Equally important is a related consideration of scheduling maintenance and replacement of the machine. For example, if during a routine maintenance operation it is known that the pump has a residual life of one hour, then it would be advantageous to replace the pump at this time. This would prevent needless down time of the machine and also tend to lessen the detrimental effect of the imminent failure on the other system components.

In essence, the criteria for mission worthiness is as follows. If the remaining life (i.e. residual life) of the system is sufficient to complete the specified mission with a known level of severity, then the machine is worthy to complete the mission. Several factors are implicit

in this criteria. It is essential to have an accurate knowledge of the remaining life, or residual life, of the system, but this residual life can only be specified in context of some known operational severity. Thus the mission worthiness can only be determined when the residual life is known in terms of the anticipated severity of the mission.

The subject of component residual life is extremely complex because of the multiplicity of factors affecting the consumption of component life and because of the probabilistic nature of component life. The evaluation of system residual life is further complicated because of the numerous interactions between the individual components constituting a system.

The life of a component or system (and thus the residual life) is typically specified from two standpoints--contaminant (wear) life and fatigue life. This report investigates the evaluation of residual life of hydraulic components operating in typical mobile systems in the context of contaminant life. An accurate knowledge of contaminant residual life when confounded with fatigue residual life provides the foundation for evaluation of mission worthiness.

This study consists of two portions: a conceptual investigation explores the complexities of evaluating residual life and proposes several methods of implementing a system for determining in the field residual life of a system. The second portion of the study performs laboratory tests in order to explore the residual life of concepts.

CHAPTER II

COMPONENT RESIDUAL LIFE

A component operated at some specified conditions will experience gradual wear of the internal surfaces resulting in degradation of the component's performance. The total length of time this component may be operated at these conditions with acceptable performance is commonly known as its Contaminant Service Life (CSL). Similarly, a component operated for a fraction of its CSL will experience some amount of internal surface wear, and it can be stated that this operation has consumed a portion of the CSL. The consumed life (T_c), stated in units of time, provides an indication of the amount of internal wear which has occurred and thus the degradation in performance. Additionally, the remaining life--residual life (T_R)--of the component is simply the difference in the CSL and the consumed life as shown in equation 2-1.

$$T_R = CSL - T_c \tag{2-1}$$

This assumes, of course, that the component is operated at constant, specified operating conditions. Unfortunately, this assumption is invalid for most mobile systems. In fact, the operating conditions (such as pressures and temperatures) will normally vary as the machine performs its intended mission, which will, of course, affect the CSL. The actual variations will be influenced by various operating factors, such as the type of mission (rough terrain or level terrain), the operator, and ambient conditions. The severity of these operating factors may be anticipated but, from a deterministic standpoint, is known only after the entire life has been consumed.

If it could be assumed that a means of quantifying the consumed life existed, then the residual life can be specified with knowledge of the operational severity which the component is to experience in the future. Although it may be possible that future operations of a vehicle are identical to past operation, such an assumption ignores the influences of varying missions, operators, etc. Thus, residual life can only be predicted utilizing estimates of anticipated operational severity.

The prediction and quantification of anticipated severity can in itself be an extremely complex task. It may be reasonable to estimate that the anticipated severity may be related to the operational severity experience in the most recent past history (say ten hours). One method of quantifying this severity is to describe the operation using cummulative data as gathered by the Statistical Analog Monitor (STAM) [1].

Residual Life Model

In order to aid in the discussion of evaluating component degradation, consider the degradation model shown in Figure 2-1. Note that the various operating conditions which the component is subjected to will result in wear within the component. This wear will typically result in dimensional changes internal to the component. Of course, these dimensional changes will then further affect the progressive wear processes. This has been represented by the Wear Model in Figure 2-1. Note that the output of the Wear Model is the complete and unique description of all internal states of the component. The complete knowledge of internal condition of the component, when combined with future operating conditions, will uniquely

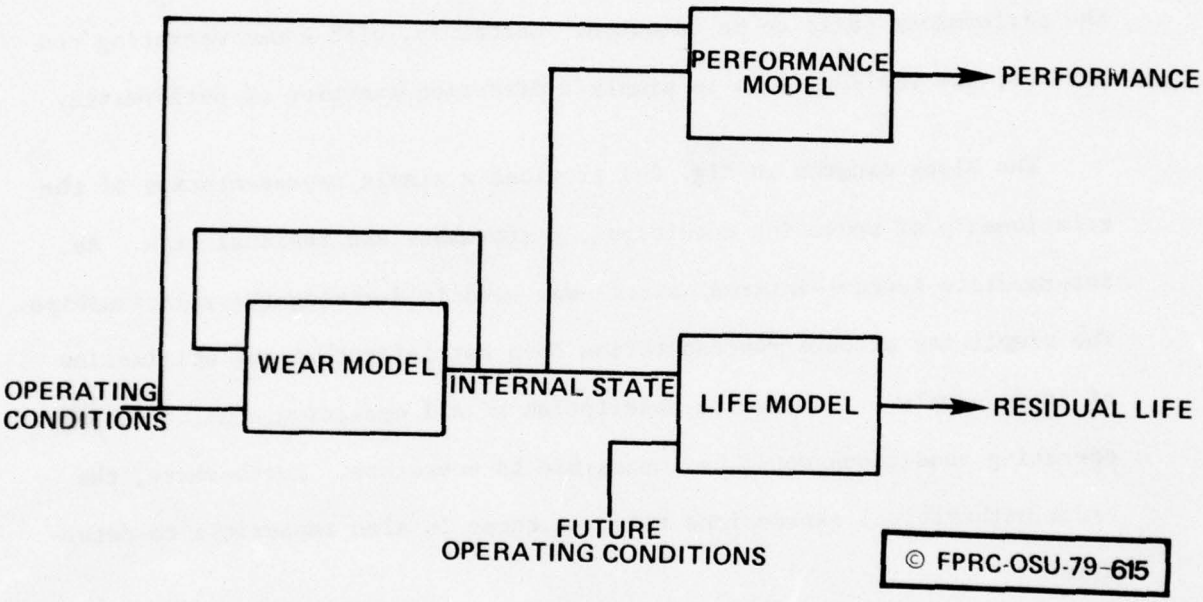


Figure 2-1. Degradation Model

determine the residual life of the component, as shown in the Residual Life Model.

Also shown in Figure 2-1 is the relationship of operating conditions to the internal state of the component and operating conditions to performance. This model illustrates the relationship betwen performance and residual life. In actuality, component life is commonly stated as a function of performance. The end of a component's useful life is defined to be when the performance fails to be adequate. Certainly, with known operating conditions, the residual life is simply a direction function of performance.

The block diagram in fig. 2-1 provides a simple representation of the relationship of operating conditions, performance and residual life. An intermediate factor--internal state--was used in deriving the relationships. The simplicity of this representation does not infer that the utilization of it is simple. The precise description of all operating conditions and operating conditions would be impossible to enumerate. Furthermore, the exact mathematical expressions relating these is also impossible to determine.

Residual Life Determination

Several approaches are reasonable for the field evaluation of residual life. These have been classified as open loop and closed loop methods. The open loop method, designated here as the Cumulative Damage Monitor, simply measures the operating condition parameters and, using the Wear Model, calculates the instantaneous damage occurring internal to the system. The total damage is accumulated providing an indication of consumed life, thus residual

life (when combined with anticipated severity). Note that this examines only the input conditions (see Figure 2-2) and much like an open loop control system assumes perfect agreement between mathematical models and system behavior. Errors and uncertainties tend to accumulate.

Another approach, considered a closed loop method, is a Static Residual Life Evaluation. This approach measures both the operating conditions and the performance and combines this information using an appropriate model to provide an estimate of the internal states. This internal state estimate is then used with the Residual Life Model for estimating residual life (see Fig. 2-3). This method is a static technique because if it is assumed that the necessary models are deterministic and unique, then the measurements may be made at any instance at time and no prior history is required for determining consumed life.

A third approach, also a closed loop method, is called the Adaptive Residual Life Estimation. This method calculates an estimate of the internal state similar to the Static Life Evaluation. This information is used in a trend analysis to project an estimate of internal state based on anticipated severity (see Figure 2-4). If it is assumed that the future operational severity is similar to that in the past, a simple projection of the curve in time will provide a time profile of the internal state. This information, when joined with the performance model will determine a time when the performance has degraded to an unacceptable level.

This last approach requires knowledge of only a model relating performance and operating conditions to the internal state, whereas the Static Life Evaluation requires both this model and a model relating internal state

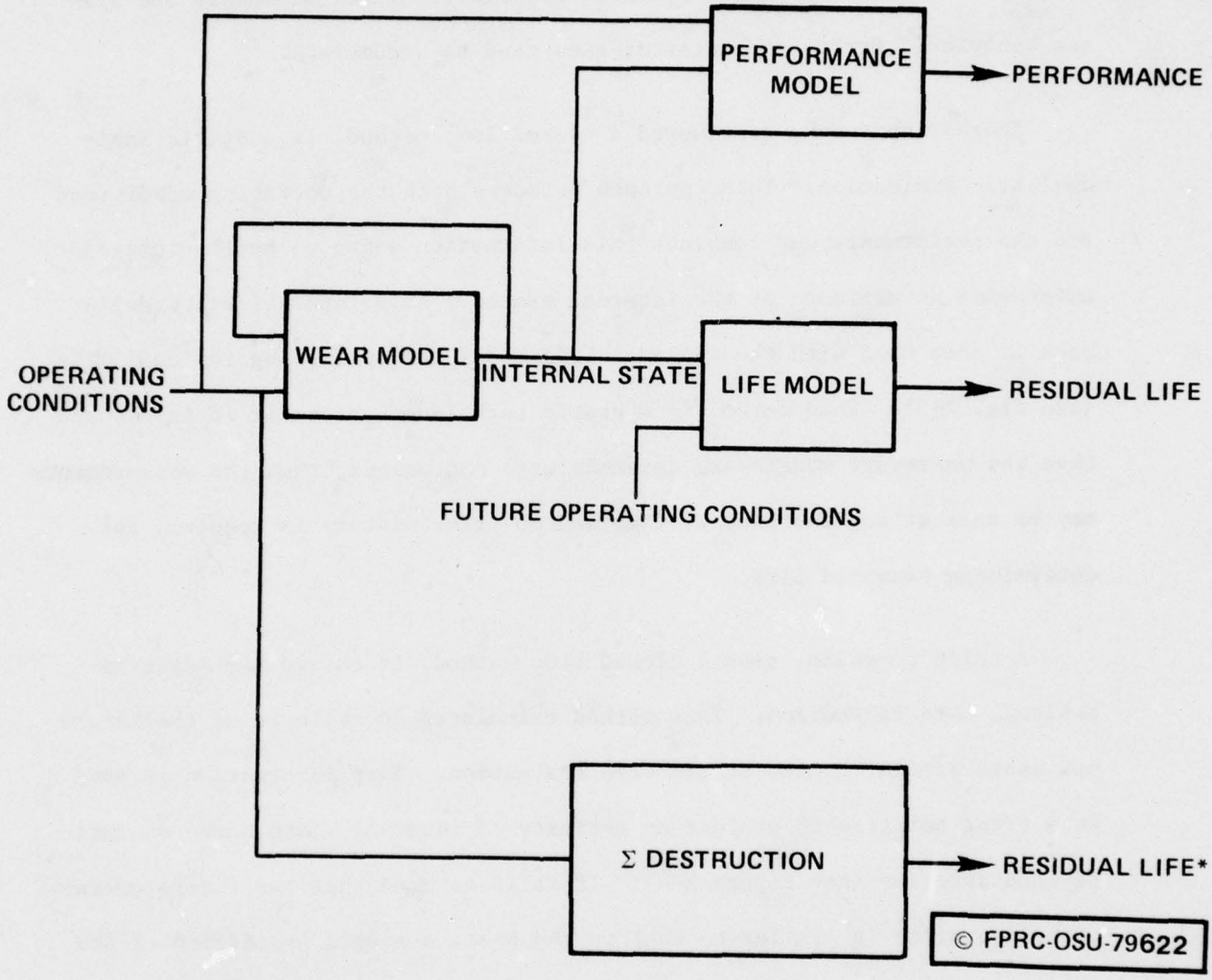


Figure 2-2. Cumulative Damage Monitor

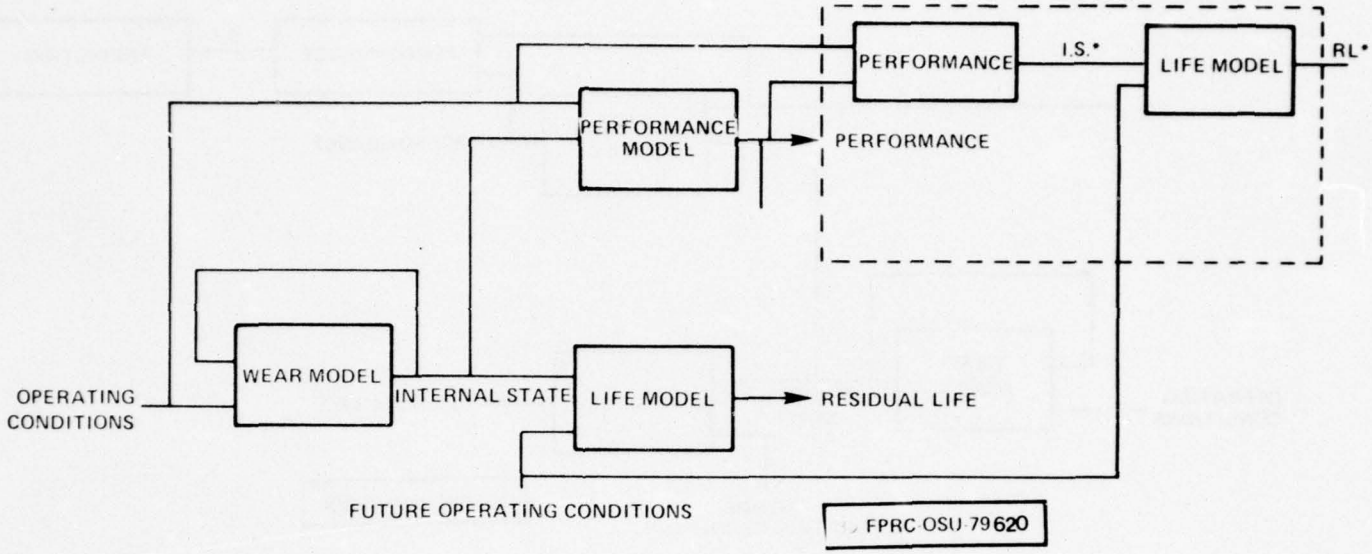


Figure 2-3. Static Residual Life Evaluation

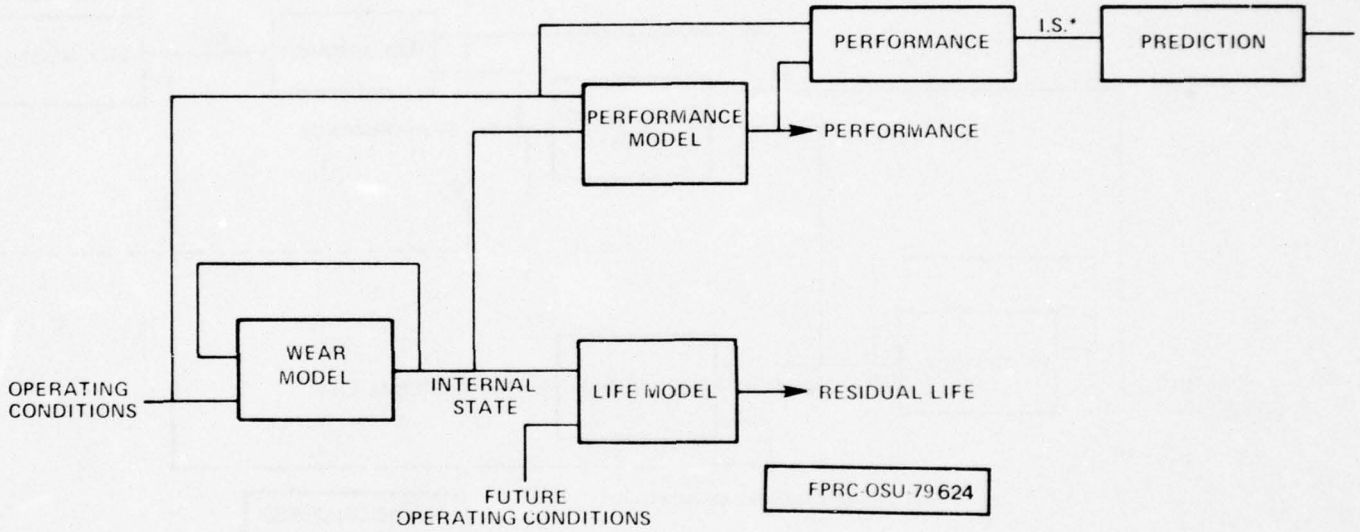


Figure 2-4. Adaptive Residual Life Estimation

and anticipated severity to residual life. Thus, the Adaptive Residual Life Estimator appears to be the desirable approach. Unfortunately, the internal states may experience little or no discernible wear during the majority of the component's life. This slow rate of change results in projection of life which is completely unrealistic.

The next chapter addresses the implementation of the Static Life Evaluation approach for use on field systems. The evaluation of residual life on a key component of the hydraulic system--a gear pump-- is considered as an example. A hardware system for implementing this device is proposed.

CHAPTER III

FIELD DETERMINATION OF RESIDUAL LIFE

The application of the residual life determination discussed in the previous chapter is extremely complex. It would be desirable to devise a device that continuously monitors the operation of a mobile system. Such a device would provide a real-time display of the estimate of the residual life of key component(s). (A similar device could be designed to be periodically connected to the system — such as during maintenance operations — to determine and display residual life.) The heart of such a device could be a single-chip microcomputer. Appropriate transducers would be installed in the hydraulic system with signal conditioning to convert this information to a form compatible with the microcomputer.

In order to examine the field determination of residual life, consider the determination of residual life of a conventional hydraulic gear pump. A simplified situation will first be considered. It will be assumed that all operating conditions except pump outlet pressure are known and remain constant throughout the life of the pump. Furthermore, assume that at some time during the life of the pump, a determination of residual life is to be performed and the outlet pressure until this time has remained constant.

Figure 3-1 shows the outlet flow versus time curves for several operating pressure typical of a hydraulic gear pump. As the pump is operated, wear occurs resulting in increased internal leakage flow. Although simplified, Equation 3-1 is commonly used to model the performance (outlet flow) of a gear pump.

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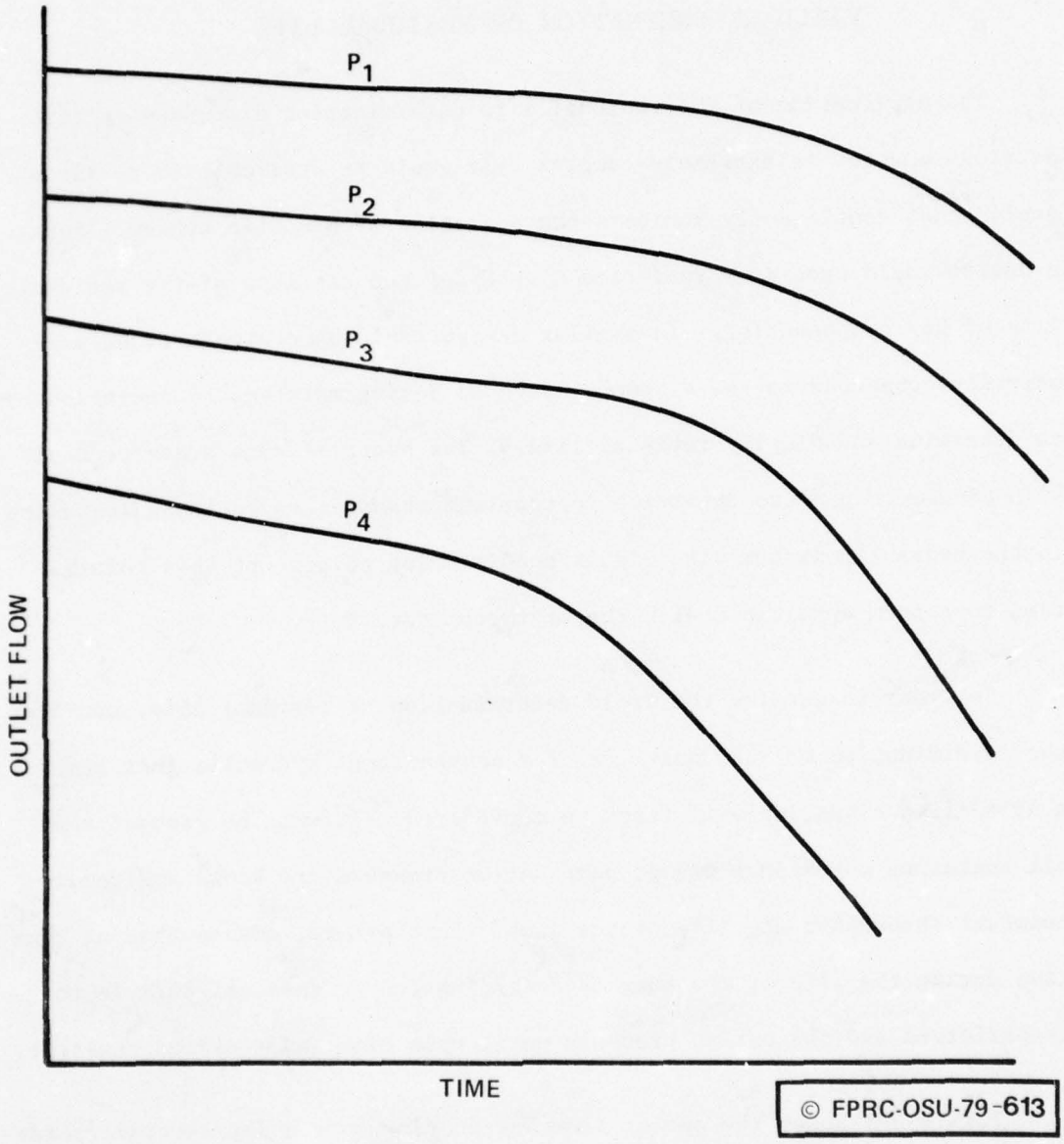


Figure 3-1. Pump Outlet Flow versus Time for Various Operation Pressures

$$Q = Q_o - \alpha P \qquad (3\text{-}1)$$

Where:

Q = Outlet flow

Q_o = Theoretical flow

P = Outlet pressure

α = Leakage flow constant

Note that the indrease in internal leakage (caused by wear) can be represented by an increase in α. Although α does not completely and uniquely describe an internal condition of the pump (such a simple description is impossible), it does provide a first approximation of this information.

The information contained in Fig. 3-1 can be represented by α vs time for various pressures as shown in Fig. 3-2. This plot can be interpreted as the model of internal state vs time for various operating conditions. Superimposed in this plot is the maximum acceptable leakage flow (Q_{LMAX}) described by Eq. 3-2.

$$\alpha P = \text{Constant} \qquad (3\text{-}2)$$

This composite plot serves as the life model.

Figure 3-3 shows a block diagram of Static Life Evaluation for the gear pump. Not that at any time (t_A), operation at a constant pressure, say P_2, will cause wear resulting in a leakage constant of α_A as shown in Fig. 3-4. The CSL for operation at P_2 is t_{LA} as shown on this figure. The consumed life is t_A hours of operation at P_2. Thus, the residual life (t_{RL}) for operation at P_2 is:

$$t_{RLA} = T_{LA} - t_A$$

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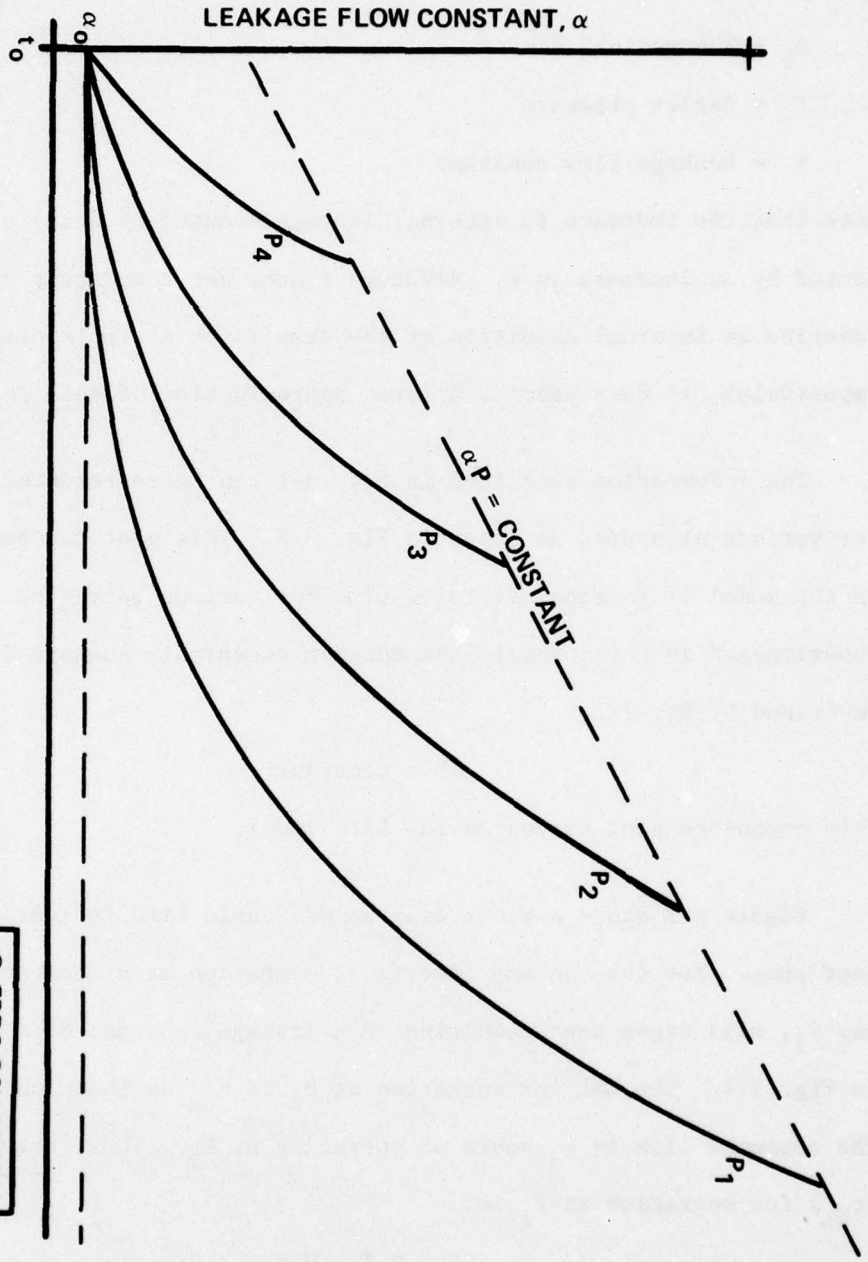


Figure 3-2. Internal State versus Time.

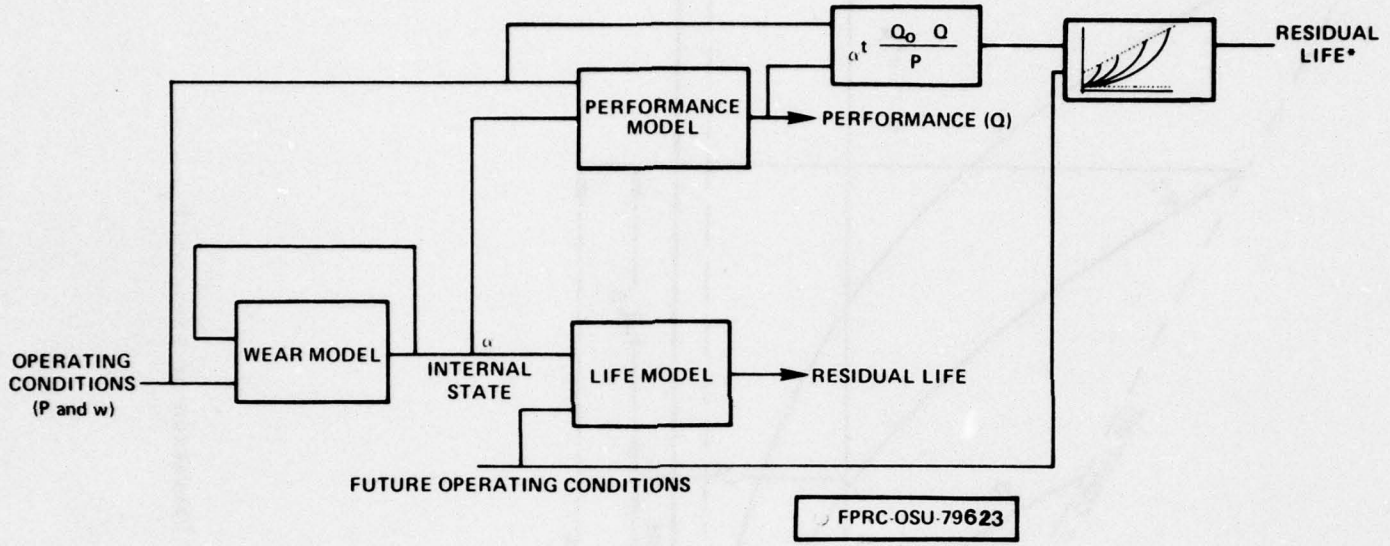


Figure 3-3. Block Diagram of a Static Residual Life Evaluation of a Gear Pump

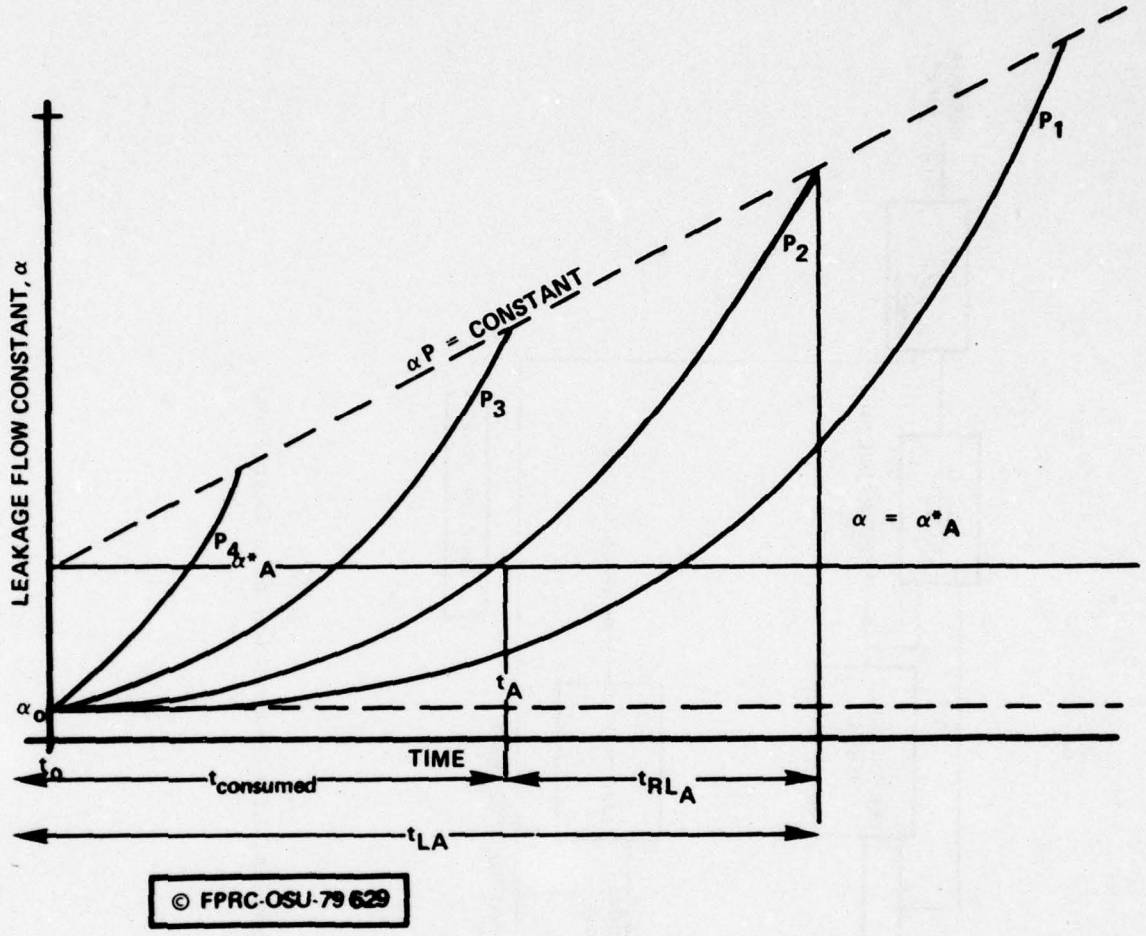


Figure 3-4. Residual Life for Operation at Pressure P_2.

Note that a change in outlet pressure from P_2 to P_3 will not result in an instantaneous change in α. Thus the consumed life can also be stated as t_B hours of operation at P_2. The CSL for operation at P_2 is:

$$t_{RLB} = t_{LA} - t_B$$

This is graphically depicted on Figure 3-5.

Thus far, determination of residual life has been very straight forward. Consider another similar situation. Suppose that all of the previous conditions are assumed except the outlet pressure will be varied as the machine performs its missions. The evaluation of consumed life can be performed as above -- measure the outlet pressure (P) and flow, calculate α, determine the time (t_c) associated with the α_3 and P_3. It can then be stated that the t_c hours of operation at P_3 has been consumed.

The residual life the component can be determined only if an accurate and complete description of future operation is known at this time. With this knowledge, the degradation of performance can be determined by accounting for the time spent at each specified pressure using the life model. For example, if it is known that the pump outlet pressure is to be as shown in Fig. 3-6, then the leakage flow constant (α_I), resulting from operating at P_4 for a time ($t_2 - t_1$) is determined. This procedure is then repeated, determining α_2 for operation for time ($t_3 - t_2$) at Pressure P_3, etc. The component life is exhausted at the time (t_n) which α is determined to be excessive. This is depicited in Fig. 3-7. Note that at time t_9, the leakage flow (αP) is excessive and thus the residual life is ($T_9 - t_1$)

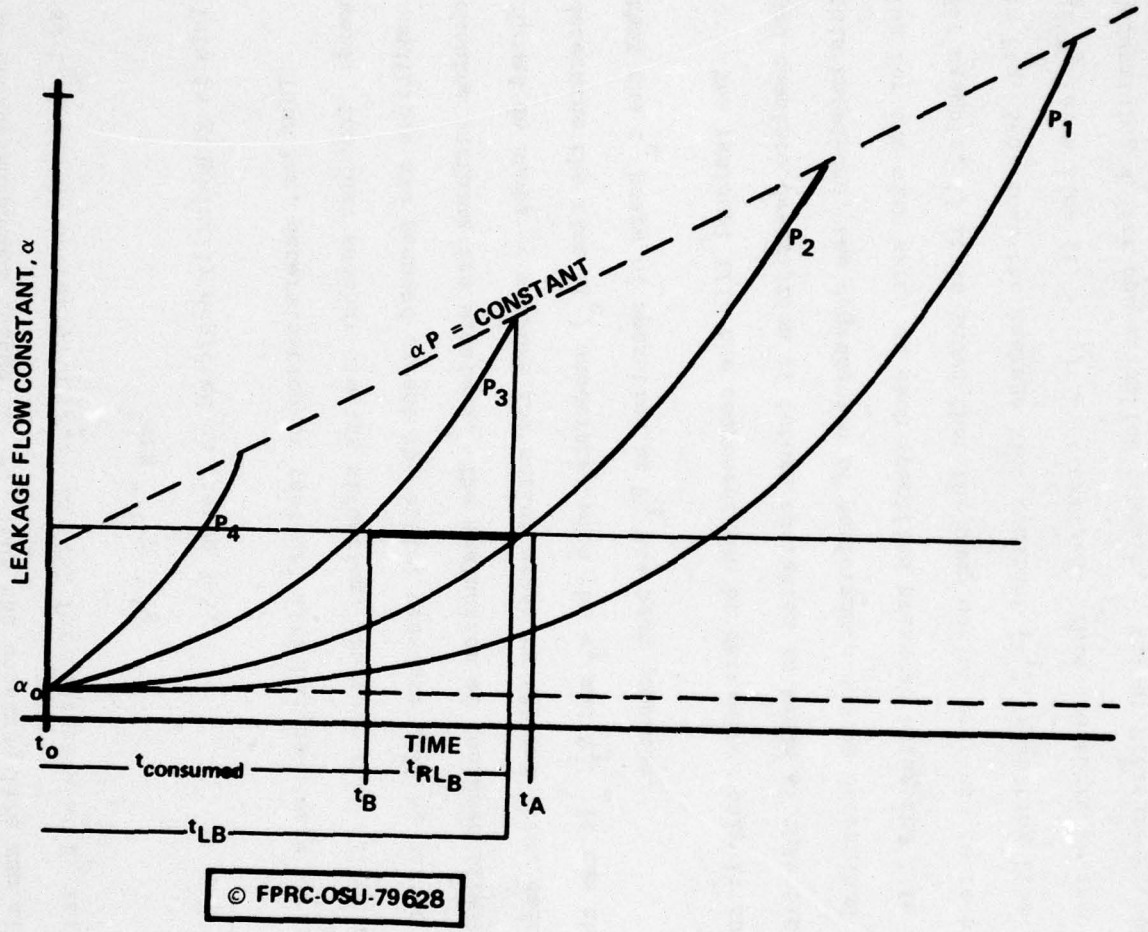
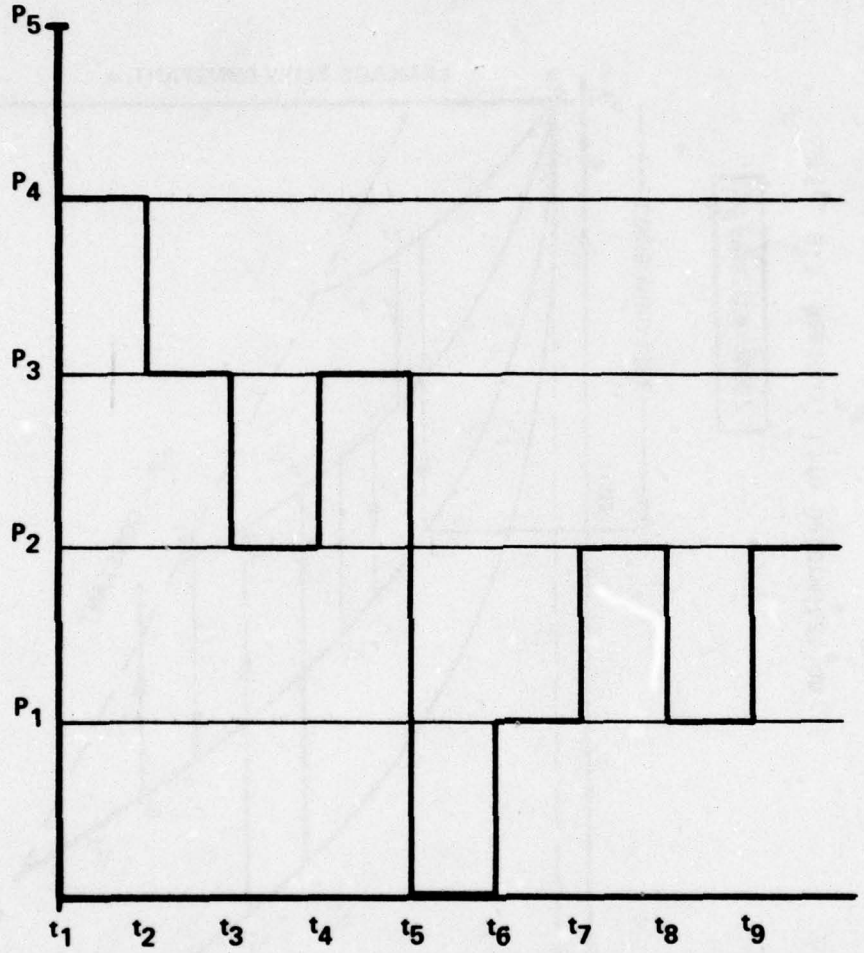
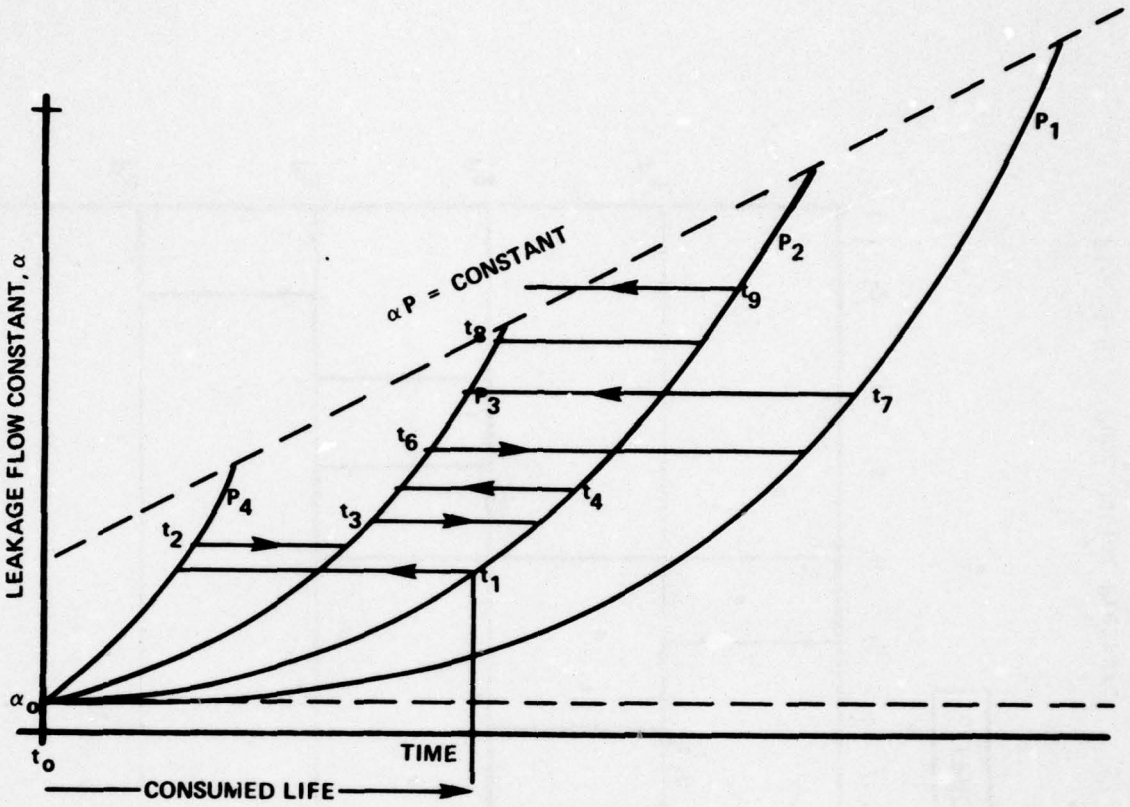


Fig. 3-5. Residual Life for Operation at Pressure P_3



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Fig. 3-6. Pump Outlet Pressure



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Fig. 3-7 Residual Life Determination

This procedure essentially performs a static measurement of operational severity (P) and performance (Q) and then simulates the future operation of the system using the life wear and life models. This process can be repeated continuously thus providing for a "continuous" indication of residual life.

It is important to note that the pressure for future time is required to perform this simulation. Although techniques have been derived which merely consider the cummulative percentage of time the system is to be operated at each pressure [2], these techniques assume a linear α versus time relationship. In general, this relationship is non-linear. (Such is the case when, say, one hour of operation at some pressure will cause more degradation in α late in the components life than in the early portion.) Thus, it is necessary to know the exact pressure — in the correct sequence— for all future time.

This is not possible for most mobile systems subjected to varying (and unknown) missions, operators, etc. Therefore, it will be necessary to estimate the future pressures. But, this may be extremely difficult. The ability to anticipate future missions (in the proper sequence) and from this infer an estimated future operating conditions will require a tremendous amount of knowledge and intelligence.

It is obvious that in order to automate the estimation of residual life, a methology must be developed for quantifying and estimating the anticipated severity. One approach which is reasonable is to assume that the future operation is to be similar to the recent past history of operation. For example, assume that the future operation will consist of

pressure excursions comparable to the last 10 hours of operation. These 10 hour excursions will be repeated until the life of the component has been exhausted.

The affect of this approach is to assume that future operations is equivalent to a short-term "average of the prior operation. Missions involving severe operations will result in an estimate of a short residual life. Similarly, missions involving mild operation will result in an estimate of a long life. The prediction will typically be changing reflecting the varying severity of the mission in which the machine is involved.

The quantitative description of the, say, 10 hours of past operation can be difficult. The storage of the pressure waveform could require an extremely large information storage capability. The Statistical Analog Monitor (STAM) can play a key role in this quantification of operating severity.

The STAM accumulates statistic on the operation of the system. The total time a parameter, such as pressure, is above a reference level and the excursions through that reference are accummulated. These statistics are acquired for eight different reference levels. (The next chapter provides a brief discussion of the STAM concept.) The resulting 16 statistics provide condensed representation of the parameter history.

A technique has been developed to construct an "equivalent" waveform using the STAM statistics [4]. The Equivalent Duty Cycle (EDC) obtained from this technique is not an exact reconstruction of the parameter waveform

but is an "average" waveform which when repeated a prescribed number of times (the equivalent cycles) identical STAM data is obtained.

Thus, the STAM can be used to quantify the recent past operation for the evaluation of residual life. The EDC can then be determined for estimation of residual life as previously described.

The next section discusses the experimental program. This effort demonstrates the application of these techniques on a controlled basis.

CHAPTER IV

EXPERIMENTAL TESTING PROGRAM

Test Procedure

The laboratory verification program was designed to duplicate in the laboratory many of the conditions found in hydraulic systems operating under field conditions. The parameters to be monitored were selected based upon previous projects of a similar nature. The test pump was exposed to a cycling outlet pressure, varying from near zero to full "system" pressure. The system was maintained at a constant volume. There was constant contaminant ingression of dispersed contaminant, through the life of the component. Filtration occurred continuously during operation of the test pump. Maintenance was performed as necessary, chainging of filters, tightening fittings, etc.

In order to vary test duration, various contamination levels were necessary. Three ways in which this can be done are:

1. Vary contaminant ingression rate.
2. Use filters with different separation efficiencies.
3. Change rate of flow through filter.

The contaminant ingression rate was varied during these tests while the filter type and flow rate were held constant.

In specific, the testing procedure utilized during this program consisted of the following:

1. Select component to be evaluated. (A pump was chosen as this is often considered the most contaminant sensitive component.)

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2. Install with appropriate inlet and outlet fittings and connections to prime mover.
3. Adjust system volume depending upon component flow rates.
4. Break in component in clean oil.

   (For pumps:) a. 15 min. at no load

   b. 15 min. at 25% load

   c. 15 min. at 50% load

   d. 15 min. at 75% load

   e. 60 min. at 100% load
5. Establish proper pressure cycle form and frequency.
6. Establish proper sampling frequency, based on expected duration.
7. Establish contamination level.
8. Commence testing and maintain:
   a. proper cycle frequency
   b. proper cycle amplitude (pressures)
   c. proper contamination level
9. Aquire data at appropriate intervals.
10. Terminate test when component is fully degraded (70% or less of original flow at full pressure, for pumps).

Test Facility Design

The endurance pump test stand schematic is presented as Figure 4-1. The pump test system was a recirculating loop consisting of a load valve, flowmeter, clean-up filter, and automatic temperature controller. The contaminant conditioning circuit included a 26 lpm pump and "conditioning

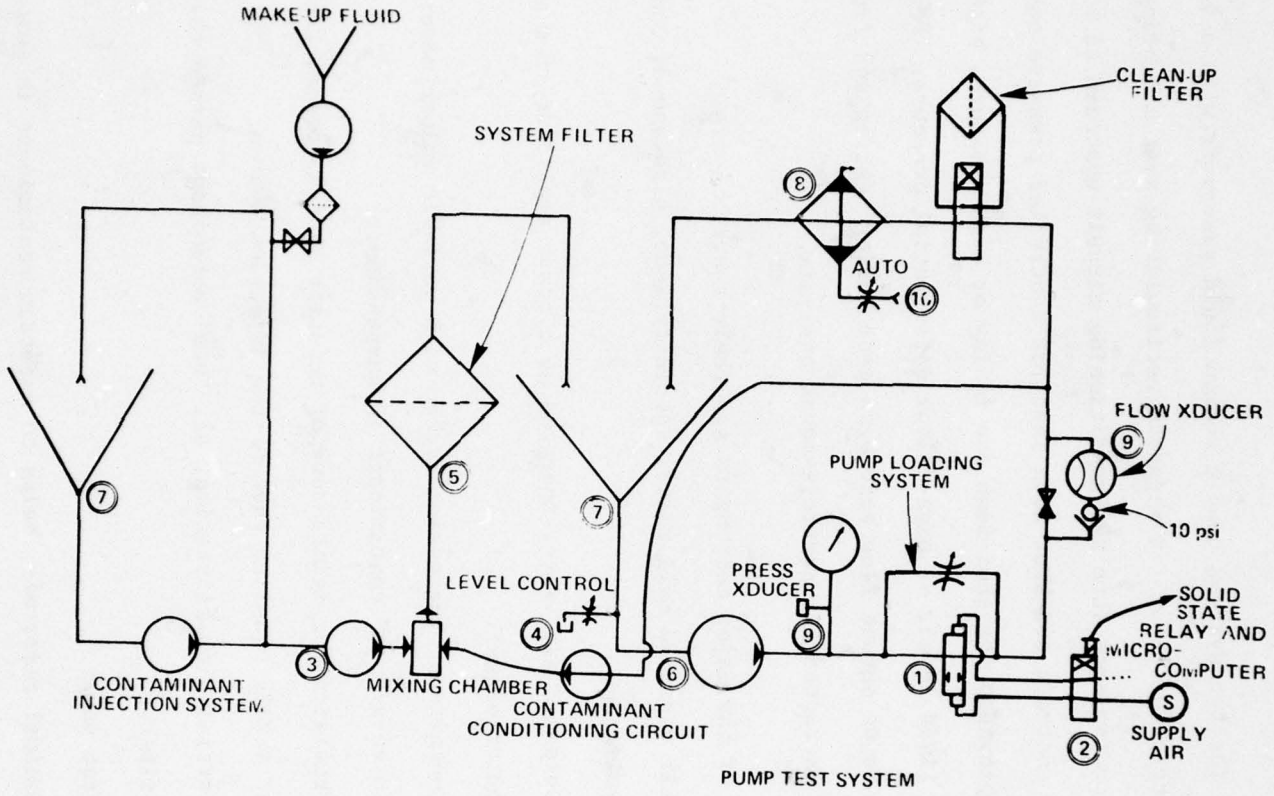


Fig. 4-1 Endurance Pump Test Stand Schematic

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filter". The two systems used a common fluid reservoir with a volume of approximately 40 litres. A separate recirculating system continuously injected contaminant into the conditioning circuit upstream of the filter. Automatic controls and shut-down features controlled pressure and temperature and provided for shut-down due to low or high pressure, high temperature, low fluid level, or high filter differential pressure. Periodic measurements of output flow rate were made during testing and fluid samples were extracted for particle-count analysis.

Some of the major components and their functions are:

1. Air operated ball valve, specially made, high speed, contaminant resistant.
2. Solenoid air valve, operated as a power amplifier from microcomputer signal.
3. Contaminant ingression system with variable volume metering pump, injects "new" contaminant in suspension.
4. Provision for maintenance of constant system volume, removes oil at a rate equal to that of the injection system.
5. Partial flow filtration, all "new" contaminant passes through filter.
6. Test pump.
7. Conical reservoir, helps to maintain contaminant in suspension.
8. Heat exchanger, removes heat created by load valve.
9. Pressure and flow transducers, output information to microcomputer -STAM unit.
10. Automatic temperature control.

Also used but not shown:

11. Microcomputer-STAM unit for automatic data aquisition.
12. Signal processing equipment for flow transducer.
13. Automatic shut down control monitoring 5 vital conditions.

Operating Conditions

All of the pumps were tested under the same conditions, except that various contaminant levels were used to obtain various pump lives.

Some of the more important conditions were:

1. 1800 RPM (Electric motor used at approximately 0.8 of rated power)
2. Flooded suction (oil level in reservoir approximately 0.6m above pump inlet, with proper suction line size).
3. MIL-L-2104 class 10 test fluid (Gulf XHD-10w).
4. 65°C operating temperature (automatically controlled).
5. <0.5 - 138 bar.
6. 3 seconds at high pressure, 1 second at lower pressure, per cycle (micro-computer controlled).
7. B80 bar/sec Pressure Rise Rate.
8. Continuous ingression of "new" contaminant.
9. Continuous Partial flow filtration to maintain contaminant level.

The traces shown in figure 4-2 are example of the pump outlet pressure and flow waveforms for this test facility.

The "Raw flow" signal was the output of the strain gauge signal processing unit, the input processed was from a type target flow meter, (10-100 lpm, Ramapo). This trace gives the output flow of the pump, downstream of

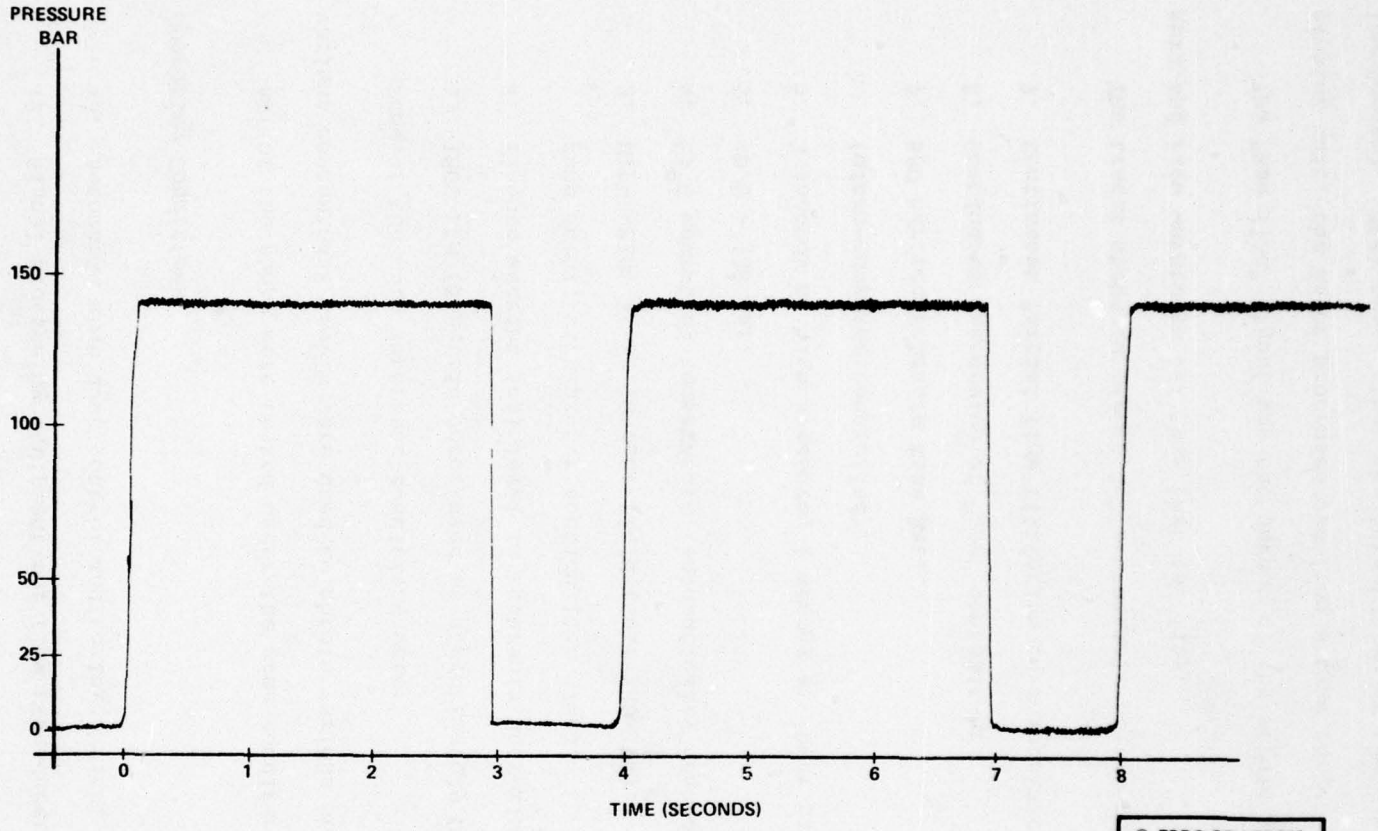


Figure 4-2A. Pump Outlet Pressure Waveform

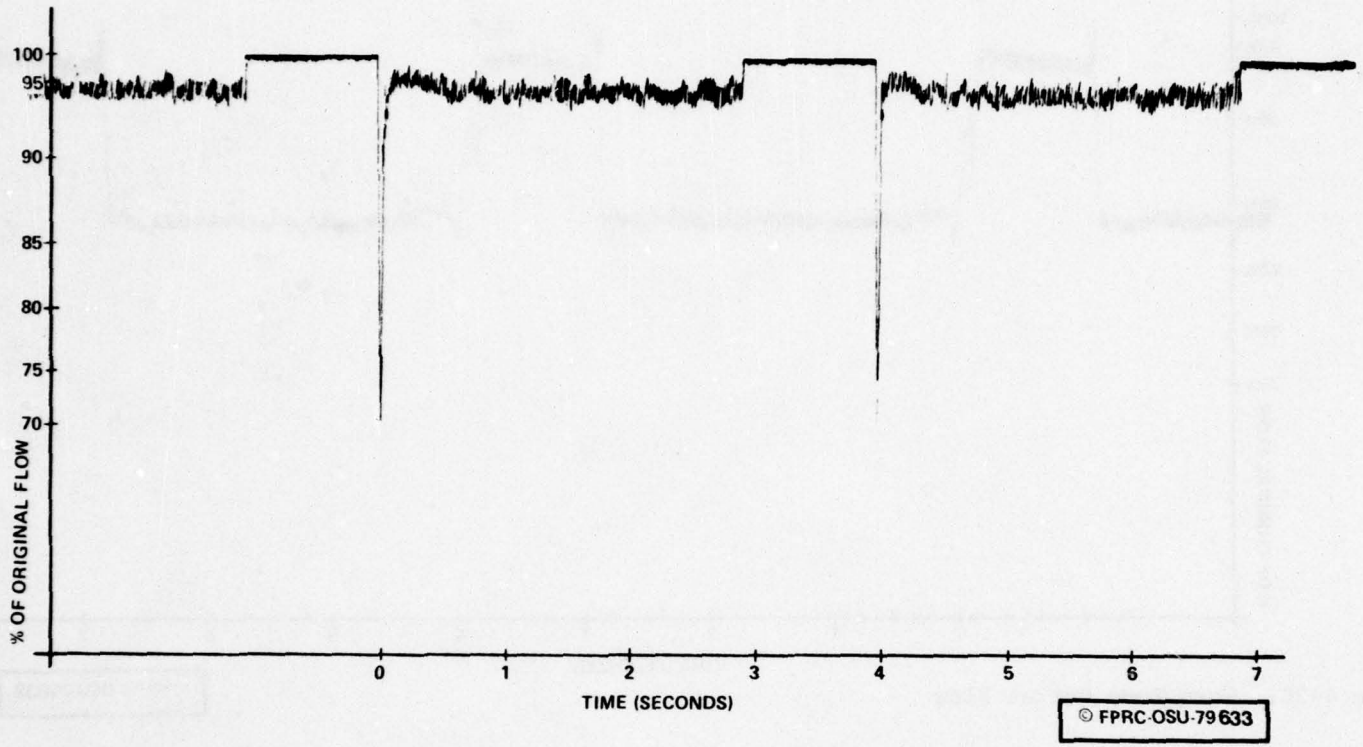


Figure 4-2B. New Pump Outlet Flow

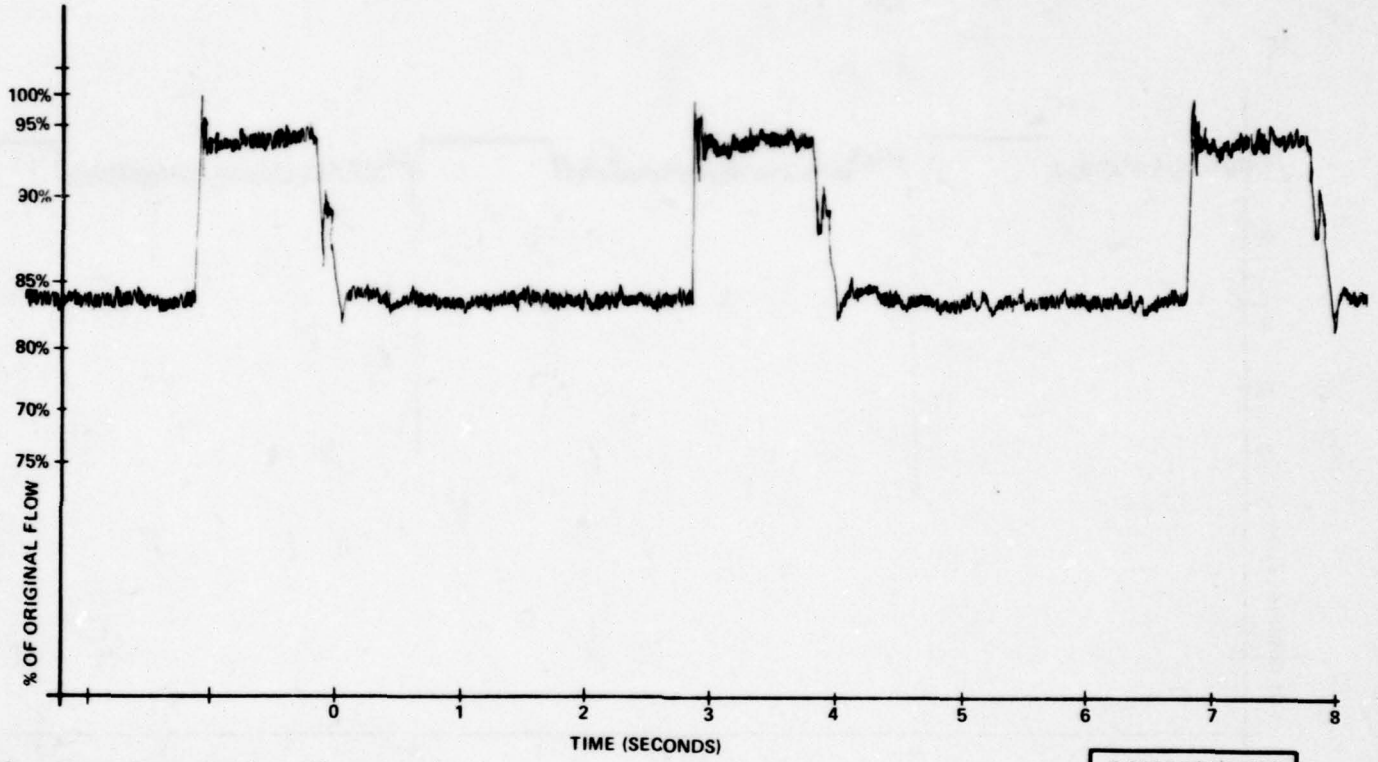


Figure 4-2C. Worn Pump Outlet Flow

the load valve. The flow "spikes" or pulses observed at the 3 and 4 second points can be attributed to fluid compressability and inerta effects, inspired by the cycle valve. The A/D Flow trace is from the input signal to the STAM unit and is the result of amplification, of setting and other electronic conditioning of the "Raw flow" signal. The pressure signal is the amplified output of a strain guage type pressure transducer. This signal was the input for the pressure portion of the STAM unit. The printed output presented is the result of the inputs shown.

Data Acquisition Techniques

In order to reduce the amount of manual data acquisition and handling, a microcomputer system was designed and fabricated to automatically acquire the STAM data. This system was designed to first sample the pressure waveform, sample the flow waveform, apply a STAM algorithm to each of these, and then print the results.

System Hardware

The hardware to perform the data acquisition is a micro-processor based system. An Intel 4040 four bit microcomputer was utilized for the data acquisition. This device performs the STAM operation, printing, and appropriate timing. An Intel 8048 microcomputer was used for providing test stand control (load cycling). Additionally, interfaces were provided for level shift and gain adjust on both input parameters. Figure 4-3 shows a block diagram representation of these two computers.

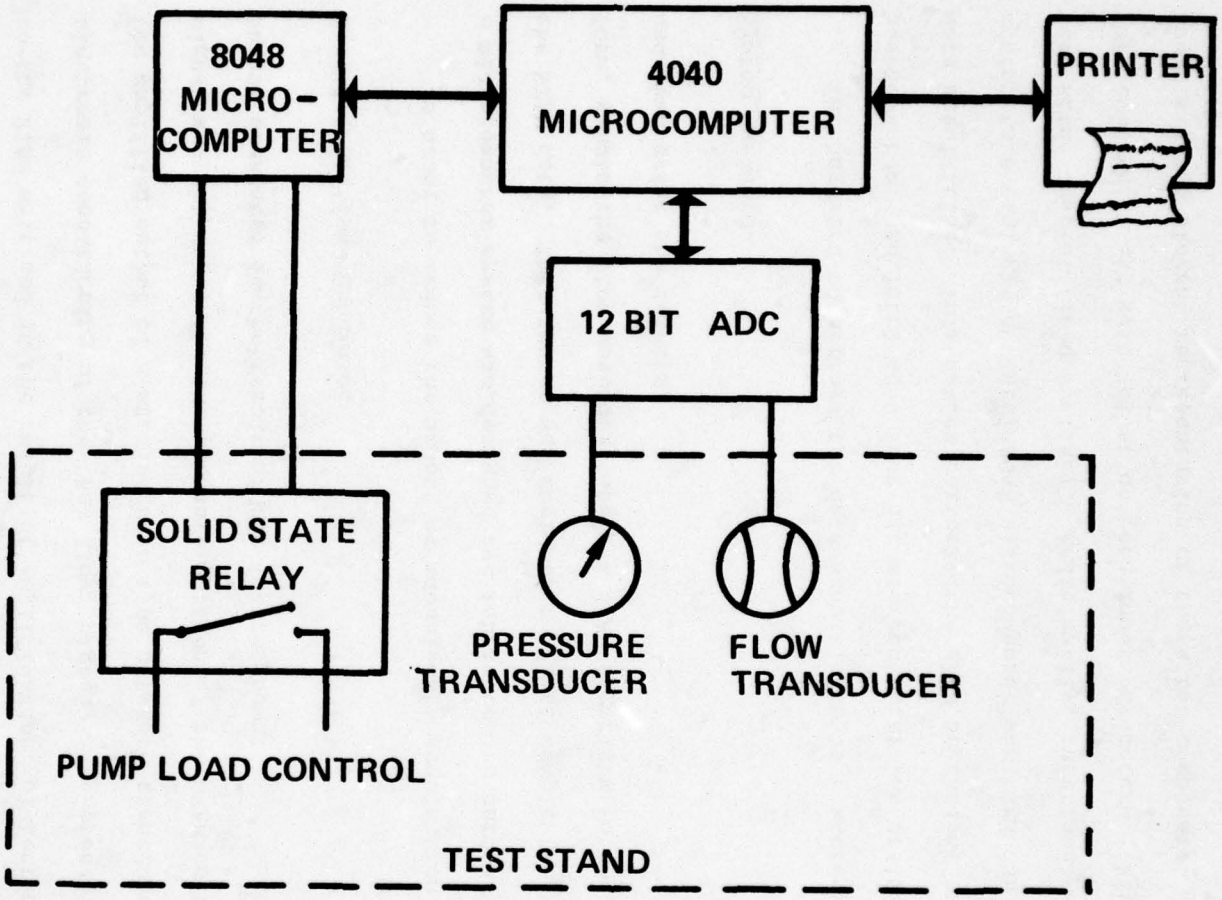


Fig. 4-3 Control and Acquisition Microcomputers

Figure 4-4 shows the basic configuration of the data acquisition system. Note that the package contains printer, display, keyboard and switches, as well as the two microcomputers.

Digital Stam Program

The digital STAM operates as follows. Upon initiation, the first parameter, pressure, is STAMed for a period of five minutes. Upon completion of this, the second pressure, flow, STAMed for five minutes also. The results are then printed on the system printer and the digital STAM enters a wait mode where no operations are performed for some specified length of time. After this time, the entire operation is repeated. Figure 4-5 shows a flow chart of the basic system operation. Figure 4-6 shows the basic time relationships of the various operations. Note that each parameter is sampled for five minutes, then printing the curves and then the system enters a wait mode. The total length of time to accomplish the entire operation is denoted as the sample period (P_s) and is varied in order to obtain approximately 100 samples for a given test.

The operations actually performed in order to STAM a parameter are as follows. The parameter is measured via the analog to digital converter (A/D). By appropriate manipulation of this measured value, a three bit number is obtained which represents the region from which the sample resides (that is, the region bounded by two reference levels). A digital counter associated with this region is incremented. If the region number has increased since the last measurements, then counters associated with the reference levels crossed are each incremented. This operation is

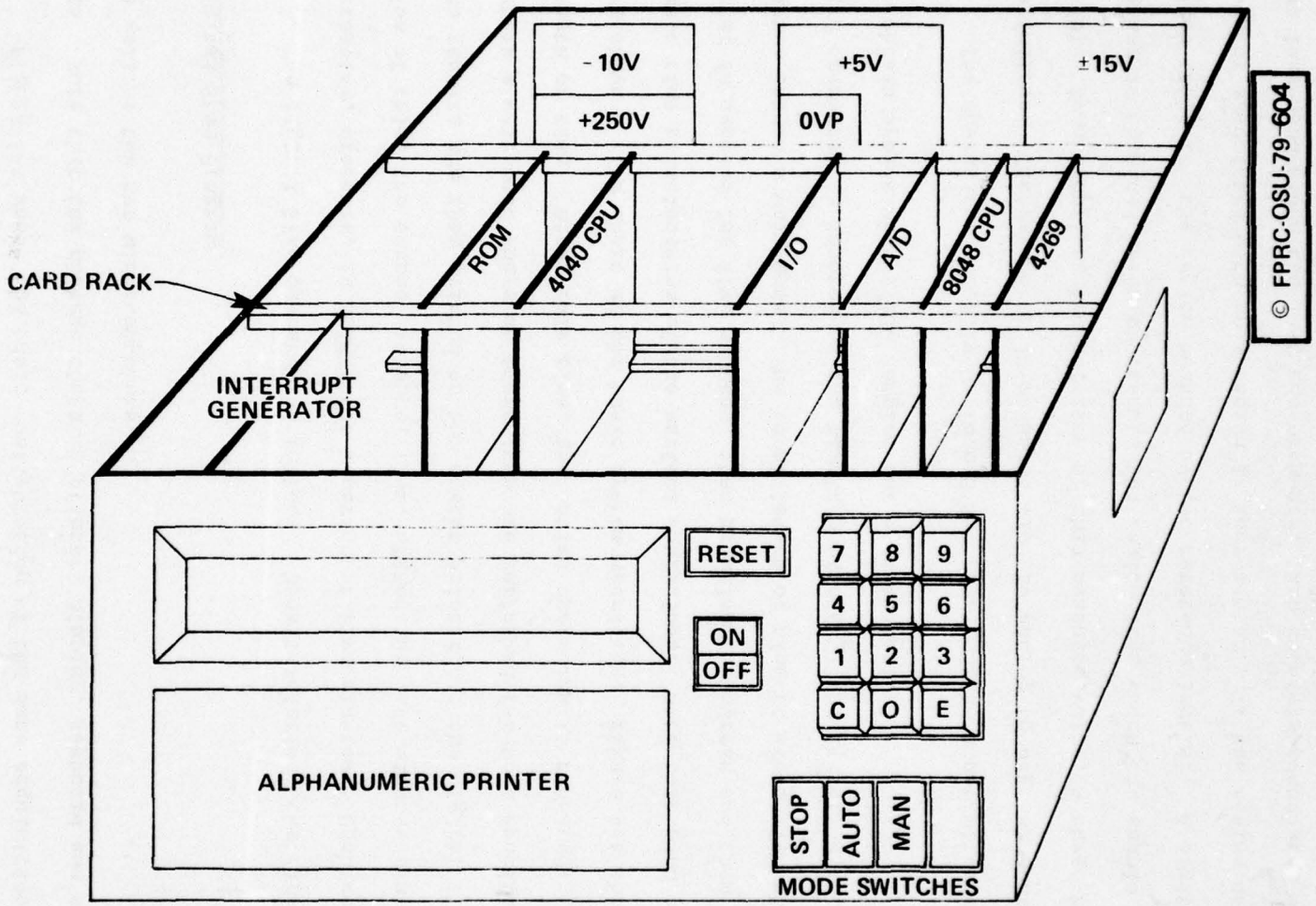


Fig. 4-4 Data Acquisition Microcomputer (Digital STAM)

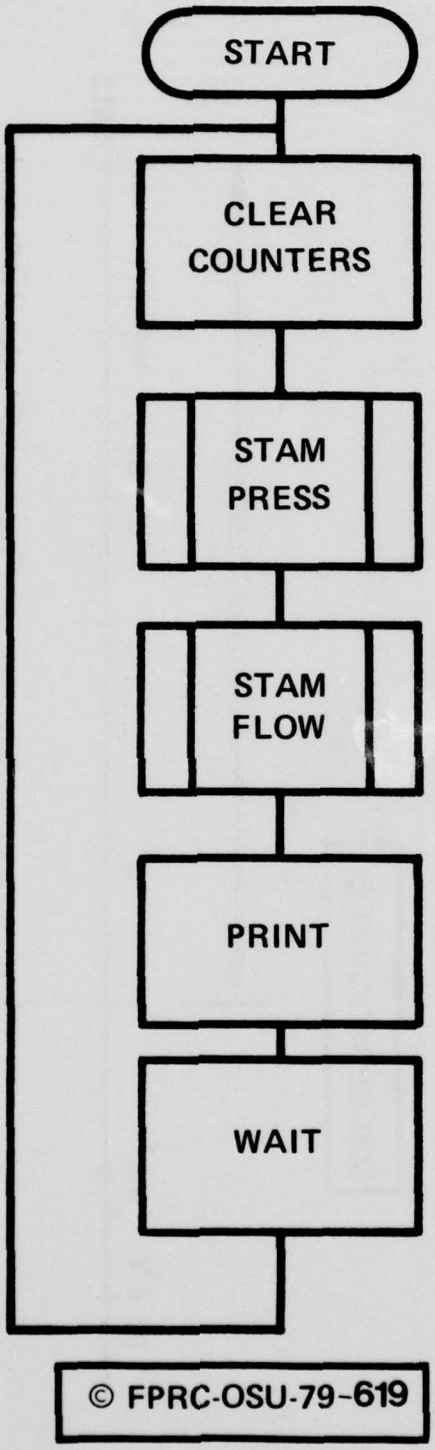


Fig. 4-5 Data Acquisition Flow Chart

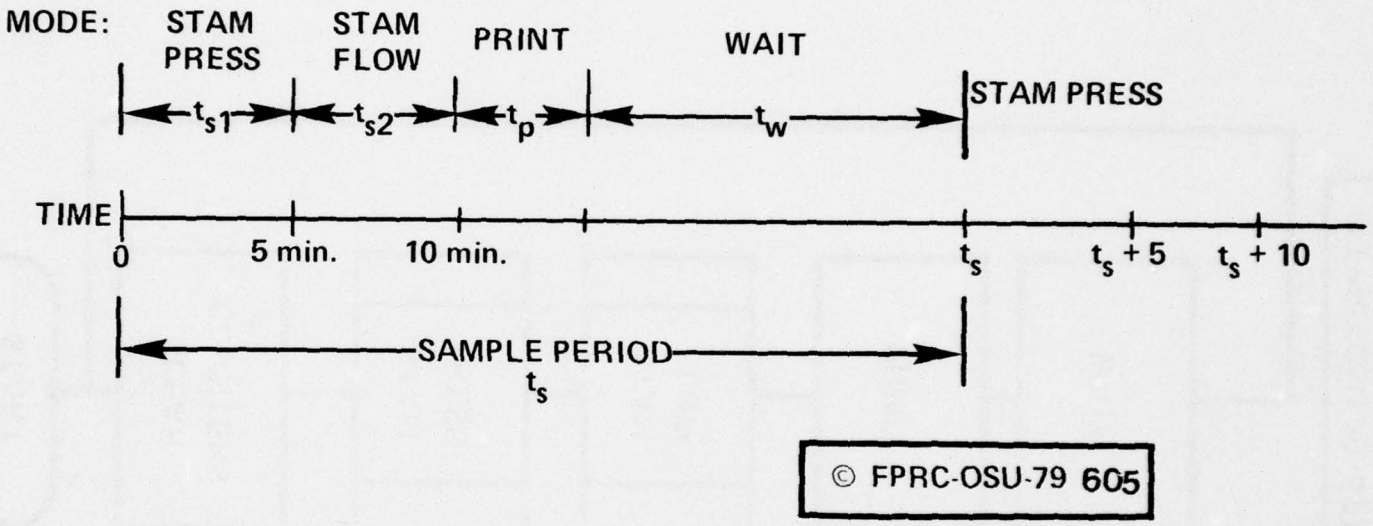


Fig. 4-6 Time Relationships

repeated every 6.362 milliseconds, thus requiring 47,169 measurements for each parameter. Fig. 4-7 shows a flowchart of this operation.

Calibration

Seven fixed reference levels have been established within the digital STAM program. These are shown in Fig. 4-8. The time in region will therefore be first the time between zero volts and reference level one, the time between reference level one and two, the time between reference two and three, etc. The level crossings will be the number of positive-going excursions through each reference level. (An additional level crossing is also acquired. It is the number of excursion into the first region.)

In order to provide flexibility, an amplifier is provided with variable gain and offset, such that the first reference level may be set and the spacing between reference levels relative to the pressure and flow information desired. For example, it is desired that the first reference level be at 20 LPM and the seventh reference level be at 60 LPM. This can be accomplished by appropriate adjustments to the scaling amplifier.

Data Reduction

Figure 4-9 shows a typical printout after a single sample for a new pump. Parameter 0 is the pressure data. Parameter 1 is flow. Channel 0 is the time between 0 parameter values and the first reference level. Channel 1 is the time between the first and second reference level. Channels 2, 3, 4, 5, 6, and 7 are similarly interpreted while channel 7 is the time between the seventh reference level and full scale.

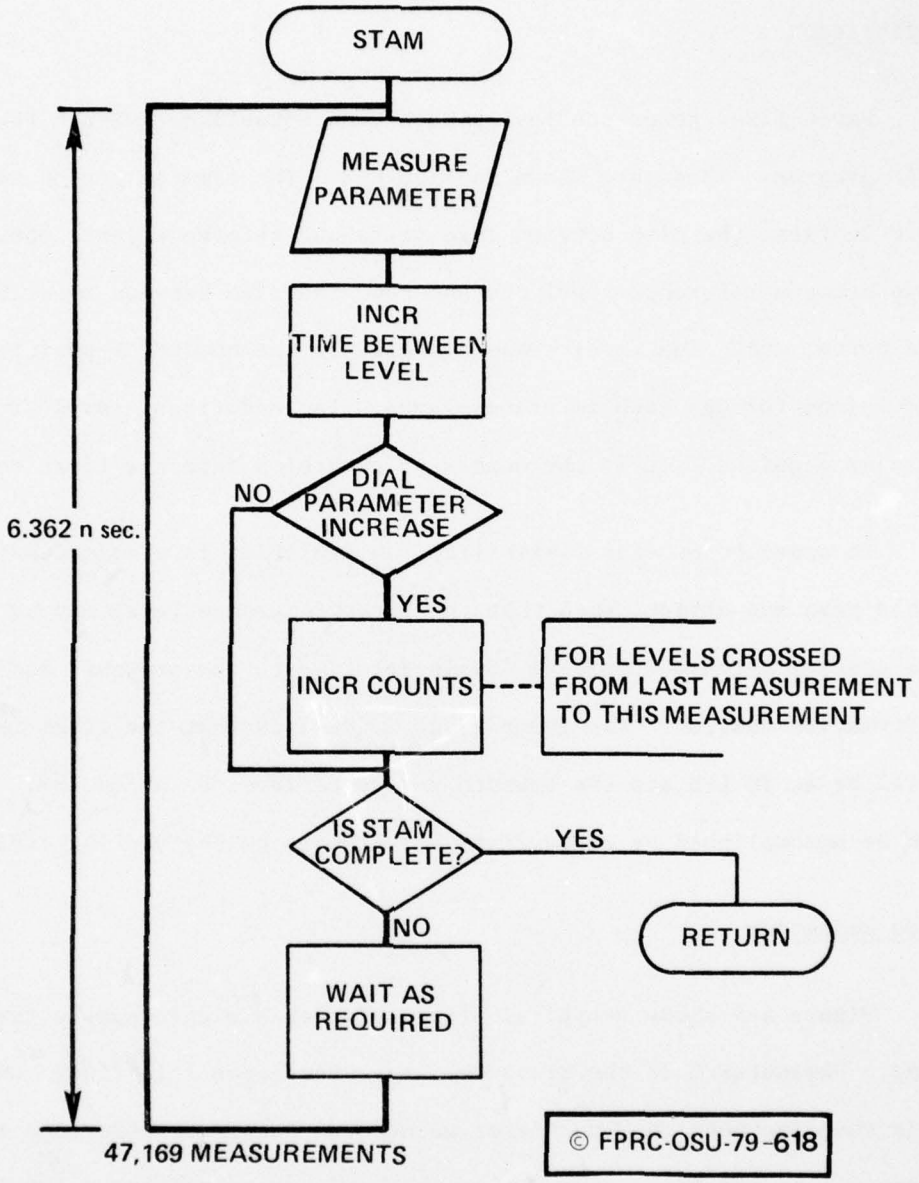
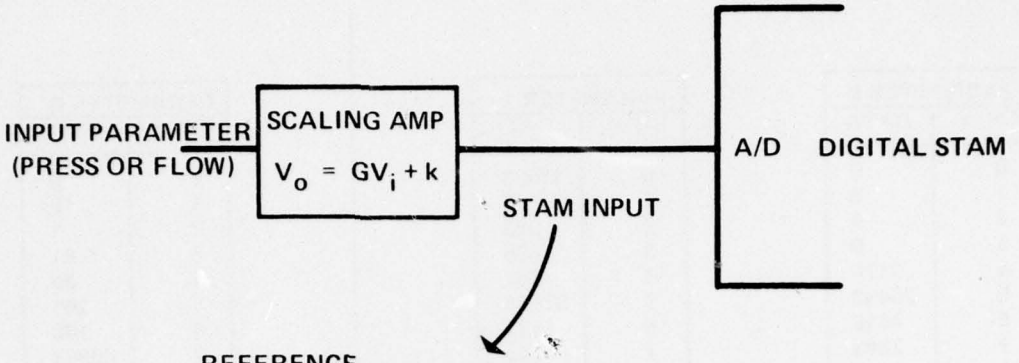


Fig. 4-7 STAM Flow Chart



REFERENCE LEVEL NUMBER	STAM INPUT VOLTAGE
1	1.248V
2	2.495V
3	3.743V
4	4.991V
5	6.239V
6	7.487V
7	8.734V

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Fig. 4-8 Reference Levels

PARAMETER 0	
CH. #	DATA
0	0
1	0
2	0
3	0
4	9170
5	25490
6	4616
7	7893
8	1
9	1
10	1
11	1
12	1
13	4295
14	142
15	1067

PARAMETER 1	
CH. #	DATA
0	11686
1	217
2	383
3	566
4	602
5	32045
6	1628
7	42
8	130
9	130
10	168
11	197
12	229
13	176
14	249
15	33

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Fig. 4-9 Typical Print for New Pump

PARAMETER Ø	
CH. #	DATA
Ø	Ø
1	Ø
2	1
3	81
4	75
5	161
6	188
7	46663
8	1
9	1
1Ø	1
11	2
12	71
13	71
14	71
15	75

PARAMETER 1	
CH. #	DATA
Ø	12462
1	128
2	9Ø
3	1Ø9
4	3Ø2
5	34Ø78
6	Ø
7	Ø
8	72
9	72
1Ø	72
11	72
12	72
13	72
14	Ø
15	Ø

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Fig. 4-10 Typical Print Out for a Worn Pump

Channel 8 is into the first region. Channel 9 is the excursions through the first reference level. Channel 10 is the excursions through the second reference level. Channels 11, 12, 13, 14, and 15 are similarly designed.

Figure 4-10 shows a typical print-out for a worn pump.

Since the time between reference levels is acquired in terms of number of counts during the sampling, this information must be converted to time using the time between measurements (T_s). This correction is performed by multiplying the counts recorded in channels 1 through 7 by 6.362 milliseconds in order to obtain the time between reference levels. The flow data (parameter 1) shown in Figure 4-9 has been reduced and is shown in Table 4-1.

CHANNEL NO.	REGION (APPROX. VOLTS)	DATA	TIME (SEC.)
0	0-1.25	11686	74.35
1	1.25-2.5	217	1.38
2	2.5-3.74	383	2.44
3	3.74-5.99	566	3.60
4	5.99-6.24	602	3.83
5	6.24-7.5	32045	203.87
6	7.5-8.73	1628	10.36
7	8.73-10	42	0.27
CHANNEL NO.	REFERENCE LEVEL	DATA	COUNTS
	All Below		
8	1.25	130	130
9	1.25	130	130
10	2.5	168	168
11	3.74	197	197
12	5.99	229	229
13	6.24	176	176
14	7.5	249	249
15	8.73	33	33

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Table 4-1

Flow STAM Data

CHAPTER V

SUMMARY AND DISCUSSION OF TEST RESULTS

As mentioned in a previous chapter, a hydraulic gear pump was chosen for laboratory illustrations of the residual life concepts. Two pump models, designated "A" and "B", chosen for the testing phase and standard contaminant sensitivity tests [4] were conducted on each model. During the only two life tests conducted on pump B, catastrophic failures occurred and no useable wear data was collected. Thus the dat for pump B is not included in this report.

Table 5-1 presents the results of the contaminant wear test on Pump A. A plot of the 1000 hour contaminant tolerance profile for this pump is shown in Fig. 5-1. This plot represents a locus of tangency points for straight line particle size destributions which would result in a 1000 hours contaminant life. The standard sensitivity test and profile are required to enable selection of an approximate contamination level to produce a desired life.

A total of three tests were conducted on pumps identical to pump A. These were labeled A1, A2, and A3. The contamination level was varied on these tests to produce both short and long contaminant lives. The average contamination levels resulting from these tests are listed in Table 5-2 and shown graphically in Fig. 5-2. It can be seen from Fig. 5-2 that all three average contamination levels cross above the 1000 hour tolerance profile for Pump A, thus less than 1000 hour lives were expected.

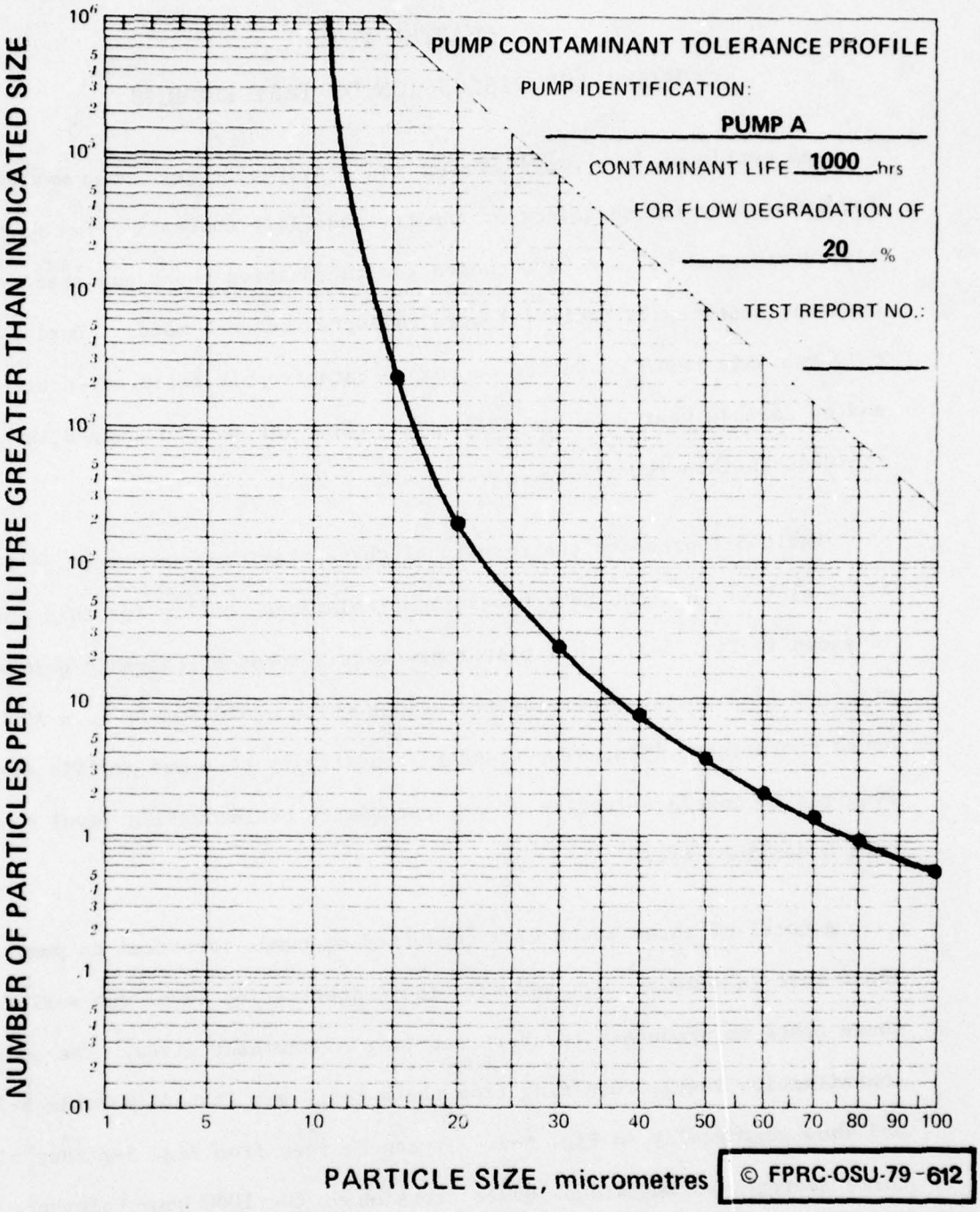


Fig. 5-1 Contaminant Tolerance Profile for Pump A

Table 5-2

Pump Test Contamination Levels

Average Particle Count/ml Greater than Designated Size

Particle Size, µm	Pump A1	Pump A2	Pump A3
3.0	928,000	2,362,000	528,000
5.0	385,000	1,090,000	164,000
10.0	73,300	192,000	11,100
20.0	2,880	7,460	256
30.0	206	380	29.7
40.0	25.6	73.3	8.87

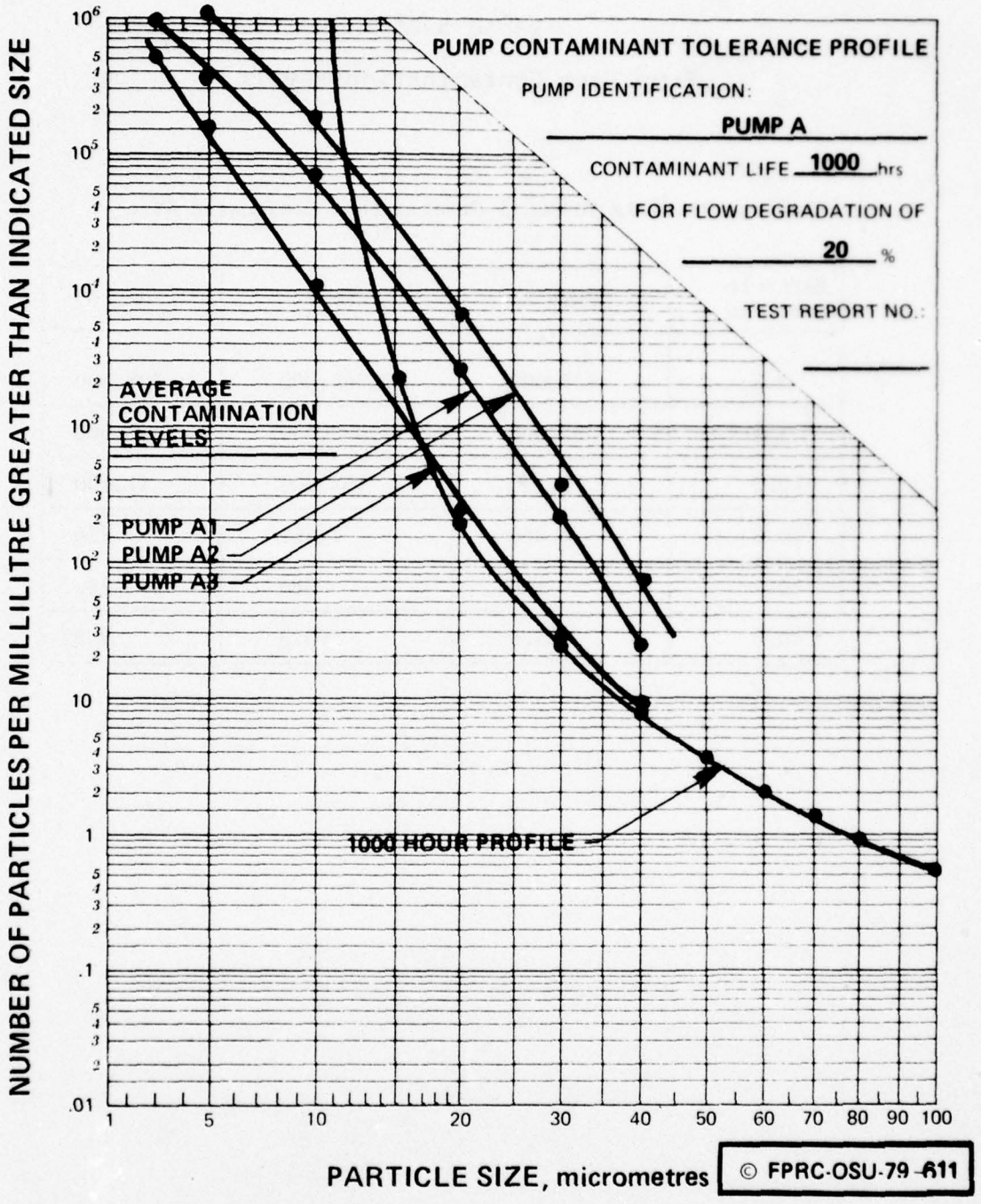


Fig. 5-2 Average Contamination Levels for tests on Pump A

The test dataresulting from the endurance tests were in the form of STAM digital output. After conversions for voltages to flow, etc., the data could be represented as shown in Fig. 5-3 for pump A3. The curves on Fig. 5-3 represent a summary of the statistical flow data resulting from STAM information taken at various points throughout the life of the pump. The curve shown at 8.3 hours represents the "flow duty cycle" at the initial stages of the test. It can be seen at this time that the flow was above the highest flow level approximately 100% of the time. The highest STAM flow level was set slightly below the initial flow of the pump in every test. After 184 hours of test time, the flow duty cycle exhibited a change in that only 25% of the time was being spent above the initial flow and 75% of the time the flow was reduced between 93% and 100% of the initial flow. This shape of curve was expected as the pump wears because of the pressure duty cycle to which the pumps were subjected. With the pump operating at high pressure 75% of the time it is expected that the flow would be reduced during that time due to internal leakage. When the pressure is low, the flow rate is again in excess of the initial except in those cases where the pump is severely worn. The remaining curves of Fig. 5-3 illustrate the change in the "flow duty cycle" throughout the remainder of the test. Similar curves were obtained for pumps A1 and A2 but are not presented here.

In order to obtain comparitive representations of the wear versus time for the various pump tests, the STAM data were interpreted in terms of the average flow rate at pressure for each test time. Because the STAM data are collected only at discrete flow levels, only an estimate of the flow at pressure for a given test time can be determined. From the curves

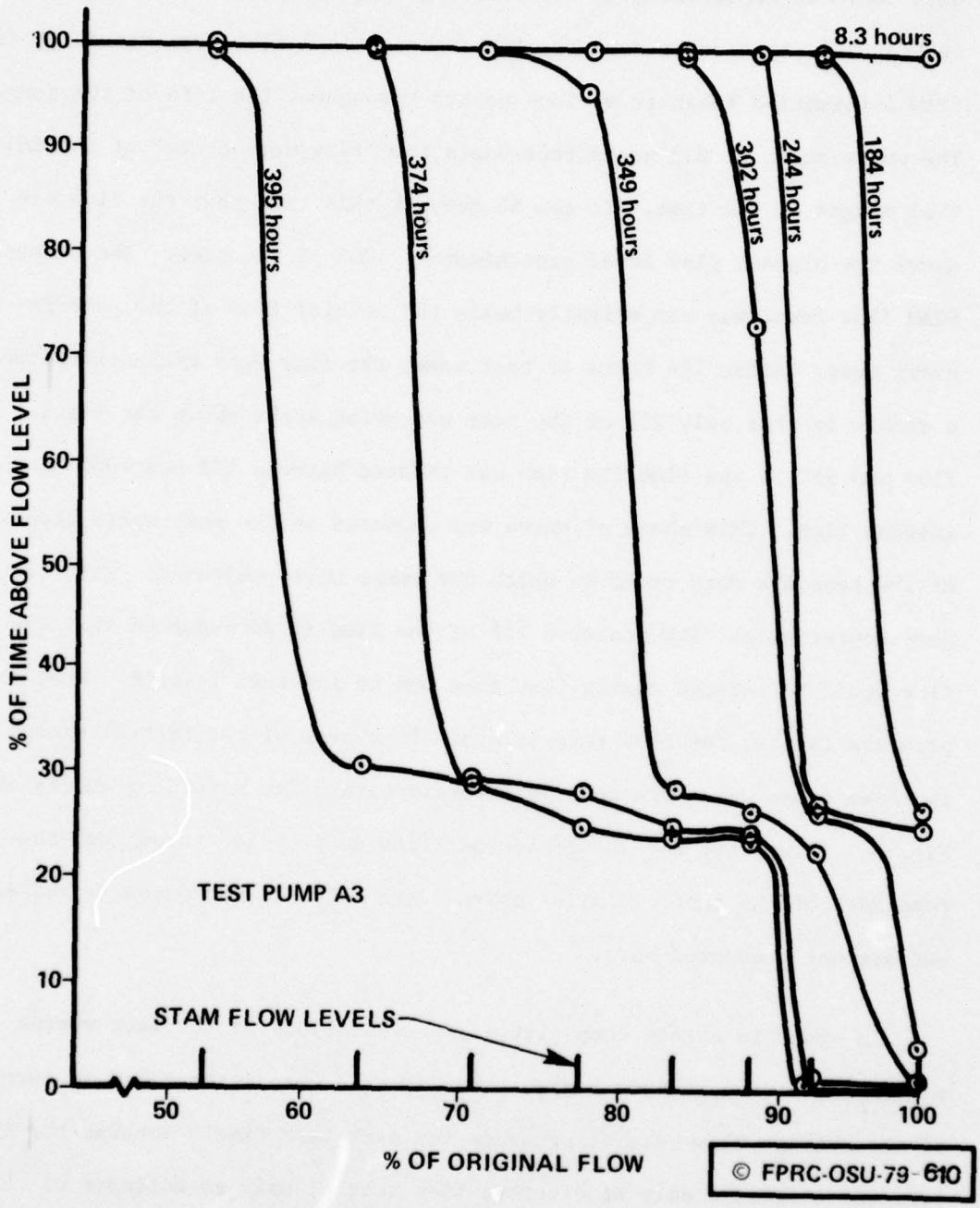


Fig. 5-3 Typical STAM Data from Endurance Tests

as typically shown in Fig. 5-3 the reference flow value is between the minimum flow value (represented by 100% time above) and the next highest value where the time above is less than 100%. For simplicity, an arithmetic average of these values was taken as the reference value. Plots of this reference flow normalized by the initial flow as a function of test time are shown in Figs. 5-4, 5-5, and 5-6 for pumps A1, A2 and A3 respectively. A smooth curve has been drawn through the data points which are discrete values due to the STAM data. It can be seen that the rate of flow degradation was different for the three pumps. If the contaminant service life of a pump is defined as the time at which the flow has degraded by 20% of the initial value, then the CSL can be read from the flow degradation ratio of 0.8. For the three pumps tested, the contaminant service life obtained is listed below.

Pump	Contaminant Service Life(CSL), Hours
A1	90
A2	53
A3	365

The relationship between pump contaminant service life and average contamination level (expressed as particles per millilitre greater than 10μm) is shown graphically in Fig. 5-7 for pump A. It can be seen that the relationship approximates a straight line on the log-log coordinates with a -0.69 slope. This differs a small amount from the value of -0.5 predicted by previous theoretical developments [5]. This deviation is most likely due to an assumption in the theory (to obtain a slope of -0.5) of a constant particle size distribution which, of course, is not true for these tests. In fact, if only pumps A1 and A2 are considered (these have

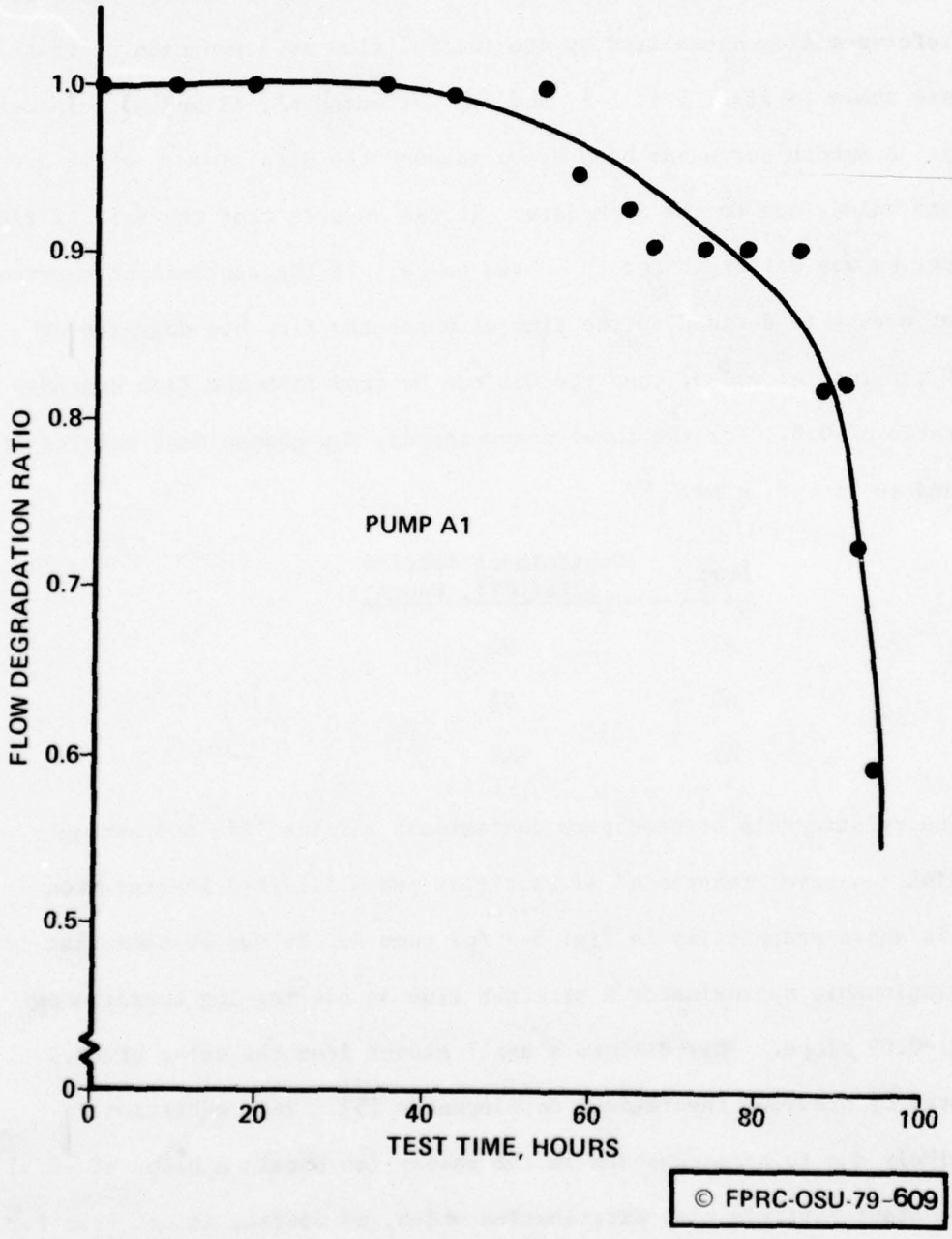


Fig. 3-4 Flow Degradation for Pump A-1

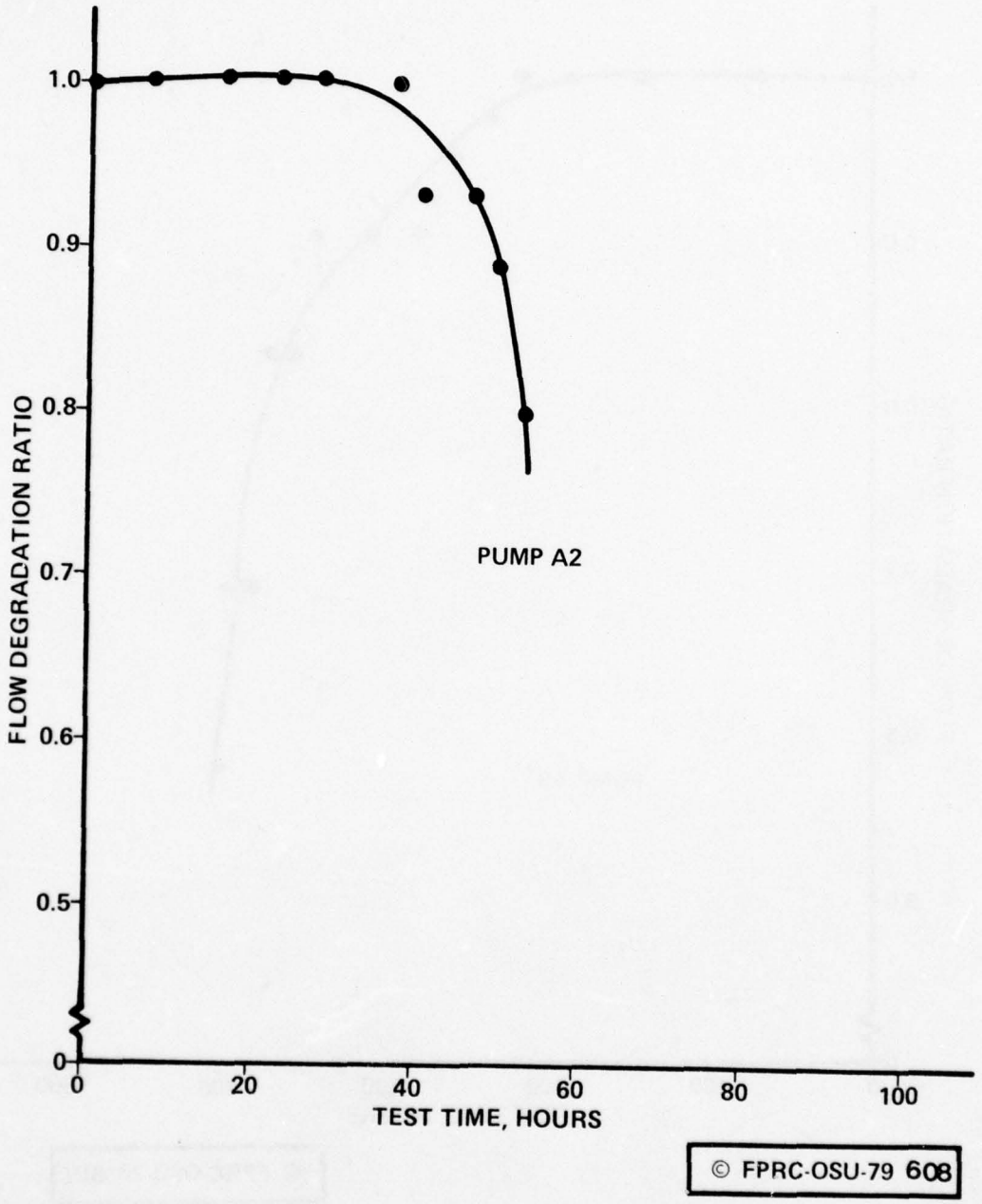


Fig. 5-5 Flow Degradation for Pump A-2

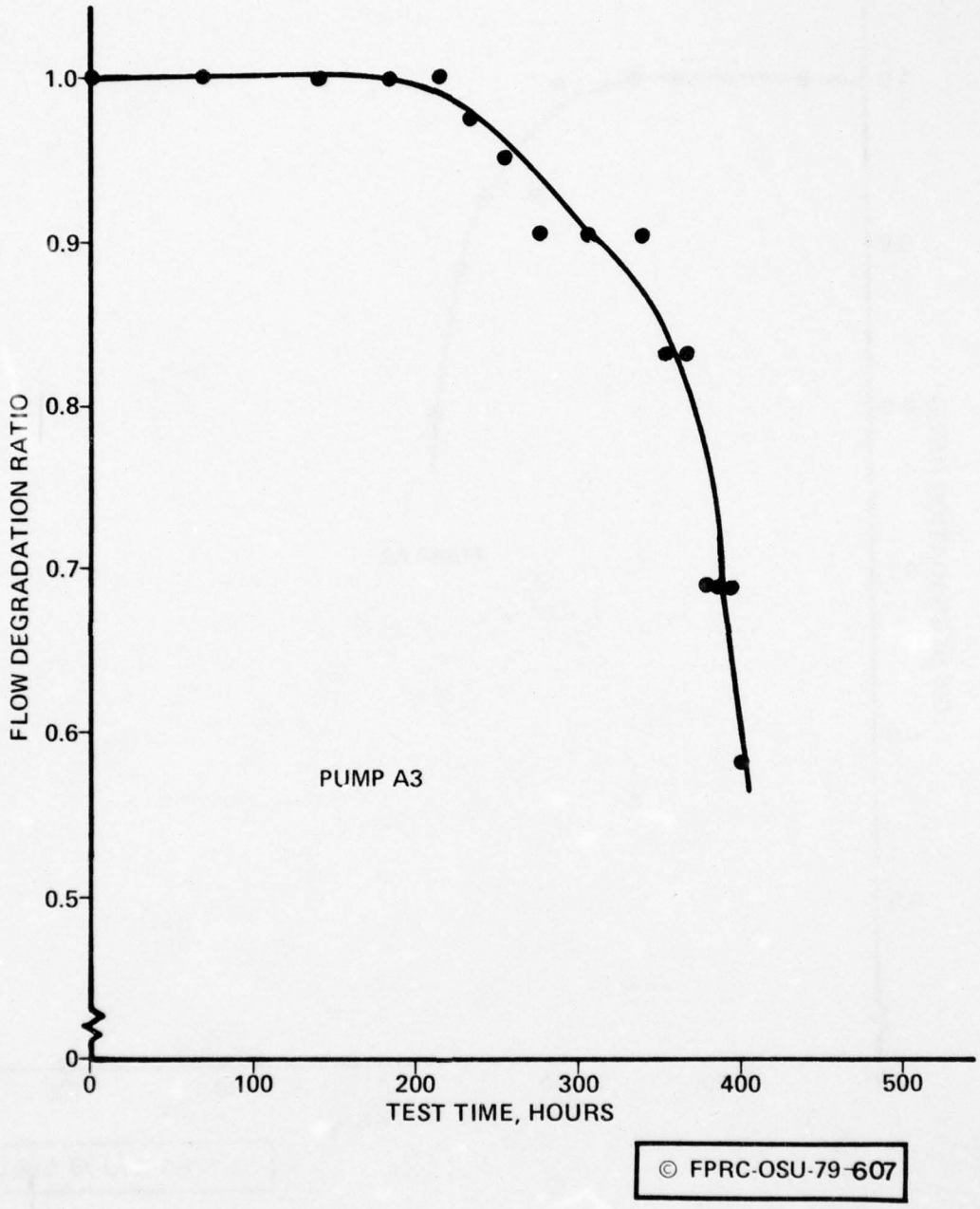


Fig. 5-6 Flow Degradation for Pump A-3

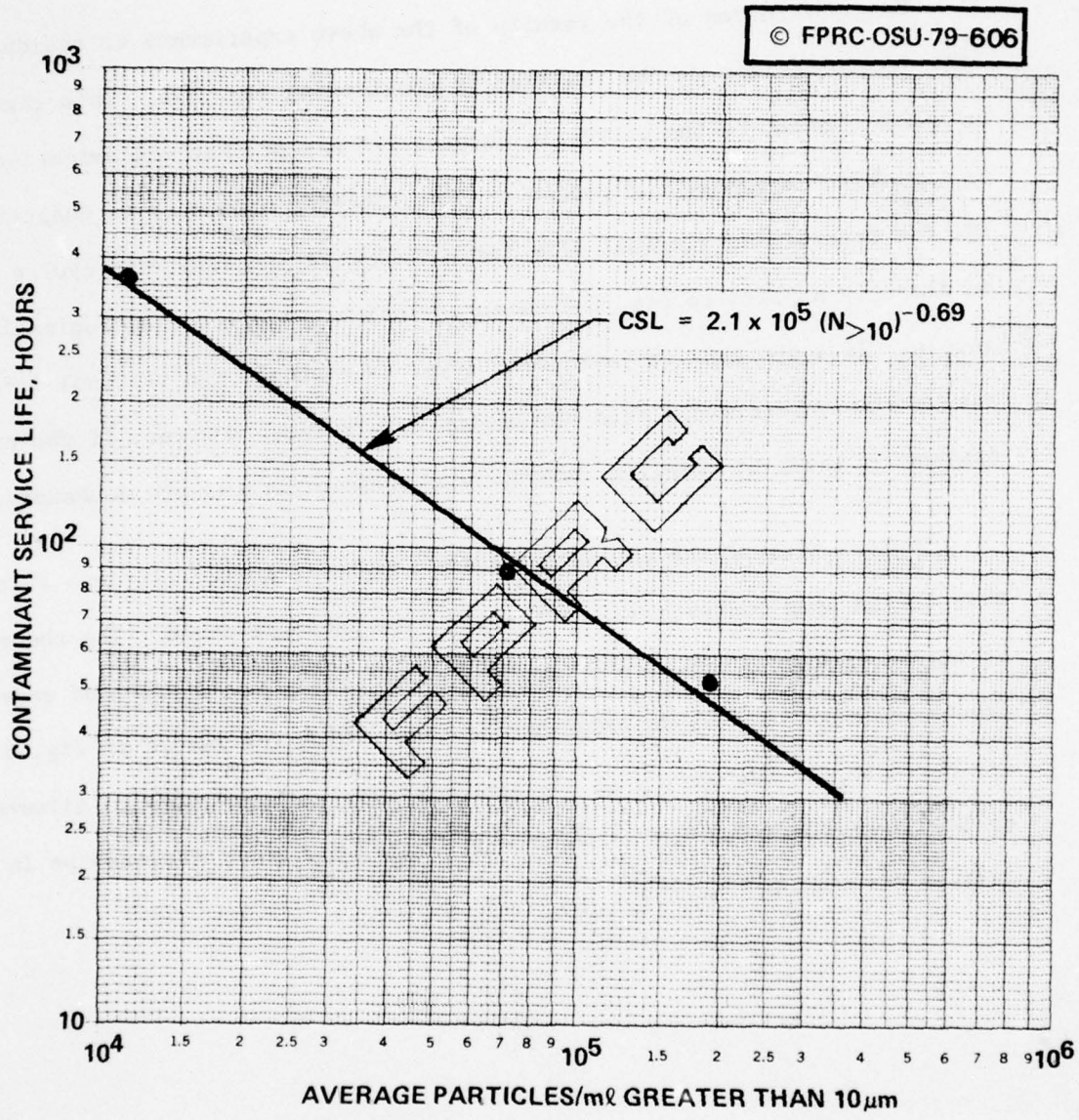
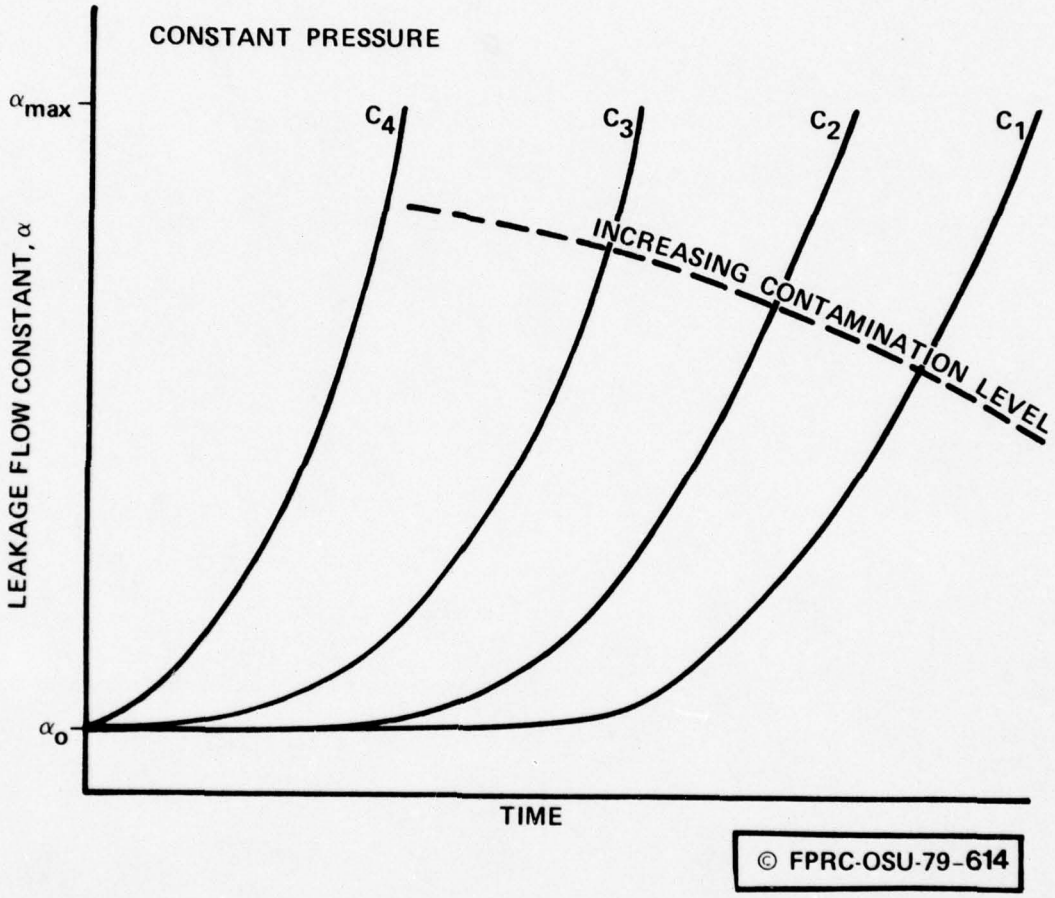


Fig. 5-7 Relationship of Contaminant Service Life and Contamination Level for Pump A

similar contaminant distribution slopes), the slope of the life versus contamination level curve would be -0.55.

The application of the results of the above experiments to residual life determinations can be realized by considering fig. 5-8. This figure is quite similar to the curves in Chapter III except that contamination is the variable instead of pressure. The concepts presented in Chapter III for determining residual life for anticipated pressure duty cycles can be directly applied to fig. 5-8 using an estimate of future contamination levels. If a constant contamination level is assumed, the residual life can be simply determined from the appropriate curve. However, if the contamination level is changing, appropriate transformations are necessary.

Construction of the curves are quite simple if sufficient data relative to the pump contaminant sensitivity have been collected. The theory developed in Ref. 5 can also be applied as verified by the previous experiments and also by past MERADCOM-OSU programs [6]. The curves in fig. 5-8 are based on the assumption of constant pressure. If pressure is allowed to vary then the situation becomes more complex and the relationships in Chapter III must be included.



Internal State vs. Time for Various Contamination Levels

Fig. 5-8

CHAPTER VI

CONCLUSIONS AND RECOMMENDATIONS

An accurate assessment of component degradation and residual life is a key factor in evaluating the mission worthiness of a mobile machine. This study has developed concepts and methodology for the determination of these parameters. It should be noted that although component degradation assessment is important in establishing the current status of a machine or system, this information must be supplemented with estimates of anticipated severity of operation in order to determine residual life.

Theoretical concepts for residual life determination for various pressure duty cycles on hydraulic pumps were developed and presented in chapter III. Such concepts were demonstrated experimentally as reported in chapter IV. The statistical analog monitor (STAM) was utilized to determine both present operational severity and performance appraisal.

Although the concepts of residual life determination were demonstrated only for hydraulic gear pumps, the basic methodology applies to all hydraulic component such as valves and motors. The descriptive equations will of course become more complex if the performance or wear model for a component is more complicated.

It is felt that this preliminary study has certainly demonstrated the feasibility of on-board residual life determination. The recently developed micro-computer based STAM [7] could readily be adapted to not only determine present operational severity but also continuously predict residual life. The primary limitation of such a concept would be in the availability

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of appropriate on-board transducers for measuring parameters such as contamination level.

It is recommended that future efforts in this area be directed first to fully verifying the concepts for a specific hydraulic pump. This would involve a complete modeling of the performance and contamination relationships followed by a thorough laboratory verification. Then, the concepts can be applied to other components and finally the total machine.

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7. Decker, R. L., "The Mock V STAM - The Application of Microcomputer Technology," The BFPR Journal, 1979, 12:265-271.

SECTION V

MATHEMATICAL MODELING AND MACHINE STATE ASSESSMENT

PROJECT PERSONNEL

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PREFACE

This section presents a detailed account of the project activities in the area of mathematical modeling and machine state assessment. It delineates the development of computer models for component and machine simulation and their use for state assessment. The work reported herein is one phase of the U.S. Army MERADCOM/OSU project on hydraulic system reliability.

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CHAPTER I

INTRODUCTION

Fluid power systems on mobile hydraulic equipment are becoming more and more sophisticated. Almost all implement control on heavy off-highway machinery such as dozers, scrapers, graders, etc., is performed by hydraulic actuators; traction in both rubber-tired and tracted vehicles is often through hydraulic drives; and steering is either hydraulically assisted or completely hydraulic. Early generations of mobile hydraulic systems relied heavily on the open-center system mostly because of its simplicity and ruggedness. Multiple actuator open-center systems can still be found on backhoes and loaders. Closed-center systems, using variable displacement pumps are gaining prominence, and leading, in some cases to integrated all-hydraulic designs. A high volume user of mobile hydraulic equipment cannot afford to overlook design trends, especially when motivated to buy commercial equipment. A perennial challenge to the equipment buyer is the drafting of meaningful test procedures and specifications for not only entire equipment but also sub-units and individual components. In order that test procedures and specifications be defensible, a reasonably good idea of the physical variables pertinent to a component needs to be obtained. Since the same piece of hardware may be used on different machines, it is highly desirable, in the interests of interchangeability, that the test procedure be component-oriented rather than machine oriented. Simulation offers a relatively inexpen-

sive method of ascertaining the test ranges for physical parameters such as pressure, flow rate and temperature. Using appropriate models, the overall efficiency and heat generation rate, and the effect of component degradation on machine performance can be evaluated.

This report presents the methodology for computerized machine assessment using empirical and semi-empirical component mathematical models. Chapter II shows how manufacturers technical specifications and test data can be used to develop component performance models. The subsequent chapter reviews computer software usable for simulating hydraulic systems. Chapter IV details the role of simulation in component state assessment and diagnostics, while Chapter V outlines the relationship between model development and verification on one hand, and test procedures on the other. Chapter VI presents conclusions and recommendations for exploiting the findings of this study. The appendix contains a library of component model forms for common hydraulic elements.

CHAPTER II

COMPUTER MODELS AND THEIR USE

Information on hydraulic components is currently being furnished in technical brochures in the form of graphs or verbal descriptions. Figure 1 presents a typical information sheet for a series of fixed displacement pumps, while Fig. 2 and 3 depict similar information on an open-center valve and a relief valve respectively. It is instructive to see how a system designer would use this information; firstly, if he was doing all calculations by hand, and secondly, if he was developing and using computer models.

FIXED DISPLACEMENT PUMP

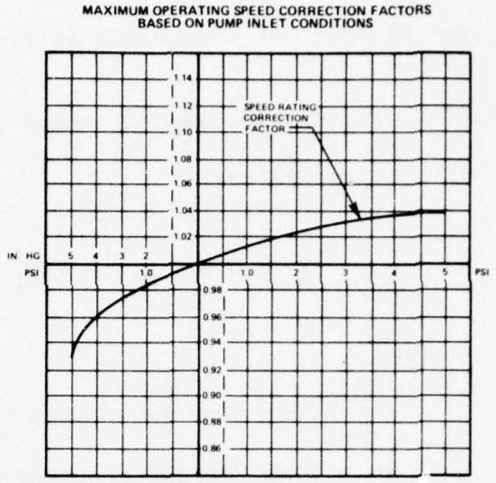
Assume that a 25 V pump with a 3.37 cu in/rev (55.23 mℓ/rev) displacement is chosen to operate between 1200 and 2400 rpm and 3000 psi (20.69 MPa). The theoretical flow rates corresponding to the two speeds are 17.51 gpm (66.26 ℓpm) and 35.01 gpm (132.5 ℓpm) respectively. The actual flow rates as read off the charts in Fig. 1(b) are 15 gpm (56.78 ℓpm) and 31 gpm (117.3 ℓpm) respectively. The volumetric efficiencies are therefore 85.7% at 1200 rpm and 88.5% at 2400 rpm. The input power can be read off the charts in Fig. 1(b) as 33 HP (24.62 kW) at 1200 rpm and 3000 psi (20.69 MPa) and 66 HP (49.24 kW) at 2400 rpm and the same pressure. The corresponding output powers can be calculated using the formula:

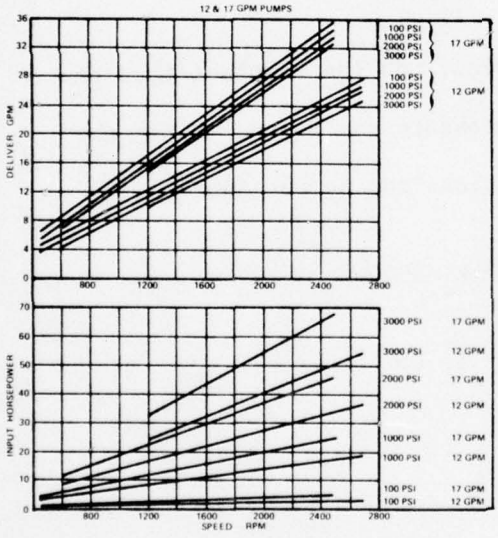
MODEL SERIES	DELIVERY GPM AT 1200 RPM 100 PSI	DISPL. CU.IN./REV.	MAX. RPM	MAX. PSI	APPROX. DEL. GPM AT MAX. SPEED & PRESSURE	APPROX. INPUT HP AT MAX. SPEED & PRESSURE	WEIGHT LBS.
25V	12 14 17 21	2.45 2.77 3.37 4.12	2700 2700 2500 2500	3000	25 28 32.5 40	55 61 68 84	32
30V	24 28	4.76 5.49	2000 2000	2000	37.5 44	52 61	34
35V	25 30 35 38	4.98 5.96 6.88 7.42	2500 2500 2400 2400	3000	47 57 64 68.5	101 121 131 141	50
45V	60 50 42	8.46 9.90 11.80	2200 2200 2200	2500	69.5 83 102	123 142 170	75

Fig. 1(a). Displacement and Pressure Specifications for a Series of Pumps.

$$H_{out} = \frac{\text{Pressure (psi) x Flow (gpm)}}{1714}$$

They are obtained as 26.25 HP (19.59 kW) at 1200 rpm and 54.26 HP (40.48 kW) at 2400 rpm respectively. Similar values can be calculated for any other operating pressure and speed. If this pump were used in an open-center system and the operational duty cycle described in terms of pressure and speed, the average power input, power output and efficiency can be calculated. The pump flow at any speed and pressure





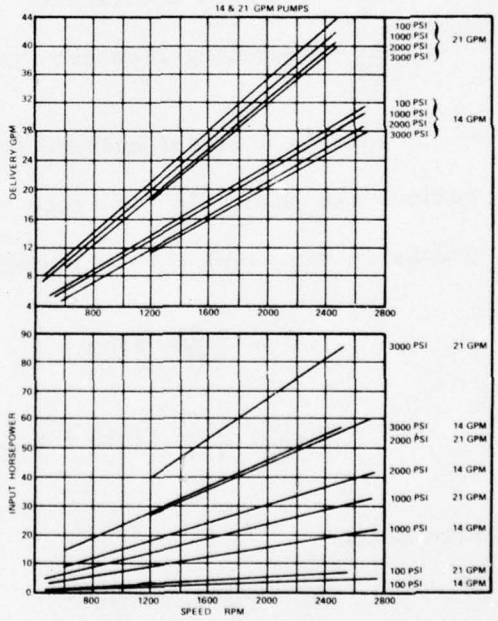


Fig. 1(b). Operational Characteristics for 25 V Series of Pumps.

can be read off the charts in Fig. 1(b).

The static computer model for a fixed displacement pump would be written as follows:

$$Q_s = D\,N + k_1 N + k_2 P_s \qquad (1)$$

where:

Q_s = Pump flow rate (gpm)

D = Pump Displacement/rev

N = Operating Speed (rpm)

k_1 = Speed Coefficient

k_2 = Pressure Coefficient

P_s = Operating Pressure (psi)

This is a linear mathematical model, and the simplest that can reflect the change in flow rate with pressure and speed. Using the graphs in Fig. 1(b) the following equations can be obtained:

$$15 = \frac{3.37}{231}\ 1200 + k_1\ 1200 + k_2\ 3000$$

$$34.5 = \frac{3.37}{231}\ 2400 + k_1\ 2400 + k_2\ 100$$

From which

$$k_2 = -762.5 \times 10^{-6}$$

$$k_1 = -182 \times 10^{-6}$$

Equation (1) thus becomes:

$$Q_s = DN - 182 \times 10^{-6} N - 762.5 \times 10^{-6} P \quad (2)$$

The power input would be written as follows:

$$H_{in} = [Q_s P_s/1714] \frac{1}{\eta} \quad (3)$$

where

H_{in} = Input power (HP)

η = Overall efficiency

Since the efficiency is a function of the operating pressure and the speed, assuming a linear relationship we can write:

$$\eta = N k_3 + P k_4 \quad (4)$$

At 3000 psi and 2400 rpm, the input and output power are 66 HP and 54.26 HP respectively. The corresponding values for 100 psi and 1200 rpm are 3 HP and 1.004 HP. Using the values of overall efficiencies obtained by dividing the output power by the input power,

$$0.822 = 2400 k_3 + 3000 k_4$$

$$0.335 = 1200 k_3 + 100 k_4$$

From which,

$$k_3 = 274.6 \times 10^{-6}$$

$$k_4 = 54.52 \times 10^{-6}$$

Equation (3) thus becomes:

$$H_{in} = \left(\frac{P_s\ Q_s}{1714}\right)\left[\frac{1}{274.6 \times 10^{-6}\ N + 54.52 \times 10^{-6}\ P_s}\right] \qquad (5)$$

Equations (2) and (5) describe the pump behavior at all pressures between 100 psi (689.7 kPa) and 3000 psi (20.69 MPa), and all speeds between 1200 and 2400 rpm.

Thus the mathematical model for the pump is:

$$Q_s = DN - 182 \times 10^{-6} - 762.5 \times 10^{-6}\ P_s \qquad (6a)$$

$$H_{in} = \frac{P_s\ Q_s}{1714}\left[\frac{1}{274.6 \times 10^{-6}\ N + 54.52 \times 10^{-6}\ P_s}\right] \qquad (6b)$$

Note that P_s and N are input quantities, which may change with time, and D is a parameter, fixed for a specific pump.

It is important to note that, subject to the limitations of a linear model, equation (6) conveys *all* the information on the 17 gpm pump, which is contained in Fig. 1(b).

Since there are at least 6 variables associated with a positive displacement pump, graphical description of performance requires two or more graphs for clear presentation. All manufacturers do not use the same quantities as independent variables. For example, one manufacturer uses pressure rather than speed for the abscissa and efficiency rather than input power for the ordinate (See Fig. 2). The total information content is, however, the same.

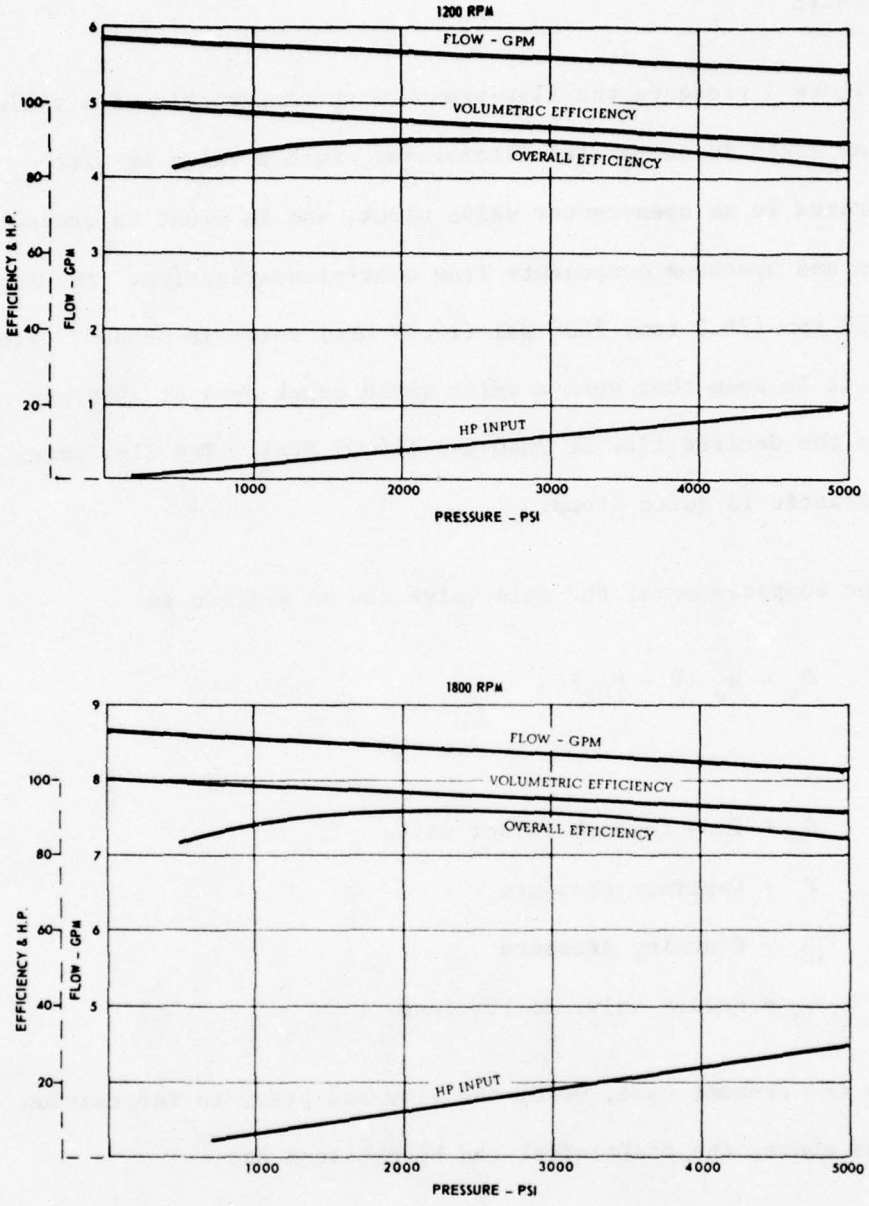


Fig. 2. Operational Characteristics of a Fixed Displacement Pump

RELIEF VALVE

Figure 3 presents the flow-pressure characteristic of a relief valve as given in commercial literature. Such a valve is often incorporated in an open-center valve block, and is meant to protect the pump and upstream components from over-pressurization. Assume that a 20 gpm (75.7 ℓpm) 2000 psi (13.79 MPa) valve is needed. From Fig. 3, it is seen that such a valve would crack open at 2000 psi and pass the desired flow at 2040 psi (14.07 MPa). The flow pressure characteristic is quite linear.

The computer model for this valve can be written as:

$$Q_v = k_v (P - P_{CR}) \tag{7}$$

where

Q_v = Flow through relief valve

P = Upstream pressure

P_{CR} = Cracking pressure

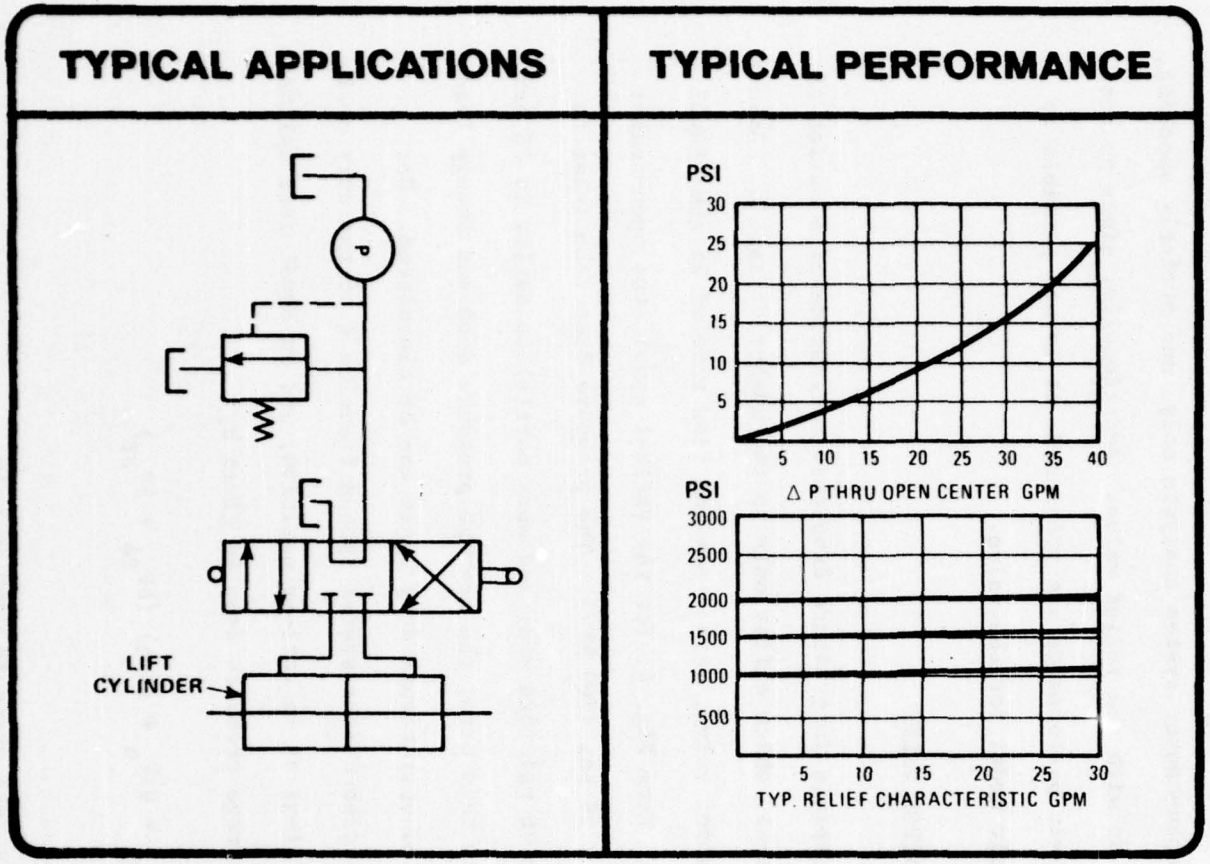
k_v = Relief valve coefficient

In the present case, using the flow and pressure information presented above, the coefficient can be obtained as:

$$k_v = \frac{20}{40} = 0.5$$

Equation (7) thus becomes:

$$Q_v = 0.5 (P - P_{CR}) \tag{8}$$



VPRP5B

Fig. 3. Typical Information Sheet on a Relief Valve.

Equation (8) is the simplest linear model for a relief valve and ignores hysteresis, viscosity effects, etc. Unless more test data is furnished it is difficult to incorporate these effects in the mathematical model. Note that the "ΔP through open-center" is pertinent for open-center system analysis only, and strictly speaking, has nothing to do with the relief valve. Additionally, since no test results of dynamic performance are presented, it is not possible to develop a dynamic model for the valve.

DIRECTIONAL CONTROL VALVE

Figure 4 depicts the pressure drop characteristics of a series of open-center valves, which differ only in the number of spools. Assume that a single spool valve, with a maximum flow rate of 25 gpm (94.63 ℓpm) is needed. From Fig. 4, for the VDP12-1 spool, the open-center pressure drop is 39 psi (269 kPa). The pressure drop from inlet to work port(s) is 76 psi (524 kPa) and work port(s) to outlet is 28 psi (193 psi). From this data, the average pressure drop and energy loss for any specified operational duty cycle can be claculated. For example, if the open-center system spends fraction q of the duty cycle in neutral and (1-q) at an extreme position, and an equal area cylinder is used, the average pressure drop is given by:

$$\Delta P_{avg} = q\Delta P_n + (1-q)\ (\Delta P_{PA} + \Delta P_{BT}) \qquad (9)$$

where

ΔP_{avg} = Average pressure drop in valve

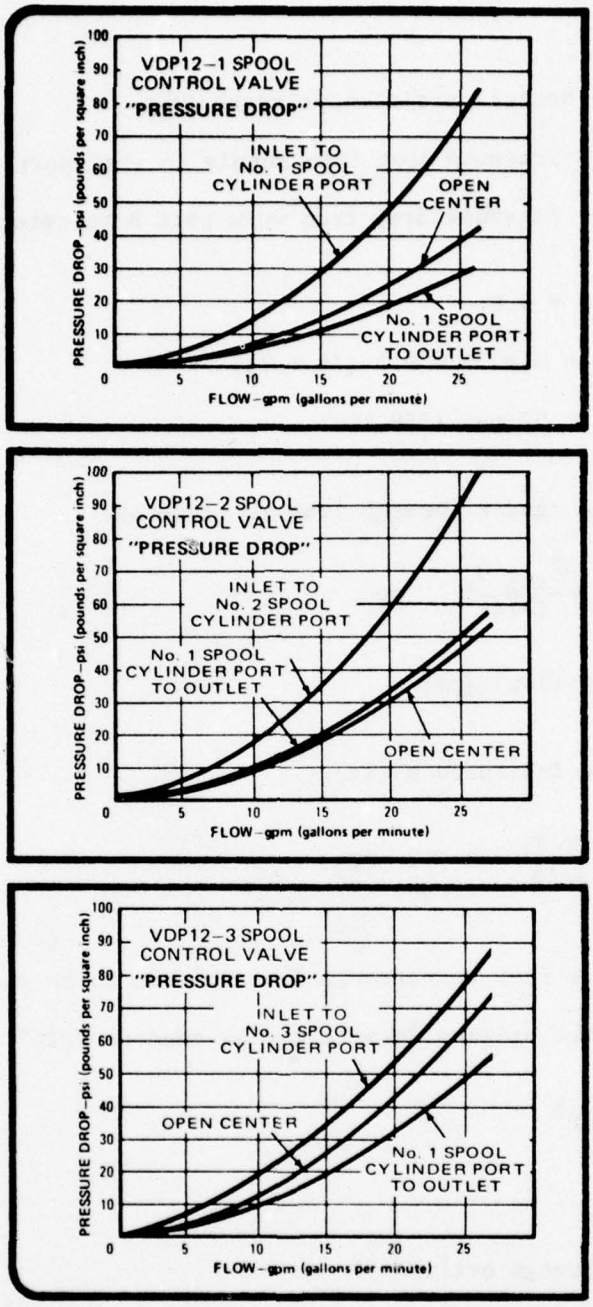


Fig. 4. Typical Specification Sheet on an Open-Center Valve

ΔP_n = Neutral position pressure drop

ΔP_{PA} = Pressure drop from supply to work port A

ΔP_{BT} = Pressure drop from work port B to return

For example, if q = 0.8,

$$\Delta P_{avg} = 0.8 \cdot 39 + 0.2\ (76 + 28)$$

$$= 52 \text{ psi } (359 \text{ kPa})$$

The average rate of energy loss is given by:

$$H_v = \frac{\Delta P_{avg}\ Q_s}{1714} \quad HP \tag{10}$$

where Q_s = supply flow (gpm)

For the flow rate indicated earlier,

$$H_v = \frac{52 \cdot 25}{1714} = 0.758 \quad HP \ (0.566 \text{ kW})$$

The pressure flow characteristics are seen to be parabobic in shape and hence the orifice formula can be used to "fit" the charts, i.e.

$$Q = k\Delta P^{\frac{1}{2}} \tag{11}$$

where

Q = flow through orifice

k = orifice coefficient incorporating the area, fluid density and discharge coefficient

ΔP = Pressure differential across orifice

For the directional control valve there are three orifices, viz, open-center, inlet to work port A, and work port B to outlet, whose coefficients will be denoted by k_n, k_a and k_b respectively. From Fig. 4, it is easy to see that:

$$25 = k_n \, 39^{\frac{1}{2}} \tag{12a}$$

$$25 = k_a 76^{\frac{1}{2}} \tag{12b}$$

$$25 = k_b 28^{\frac{1}{2}} \tag{12c}$$

or

$$k_n = 4.003$$

$$k_a = 2.868$$

$$k_b = 4.725$$

These three parameters define the valve uniquely and can be used in the computer model. Note that the general formula, equation (11) can be used in conjunction with the above numerical values, to calculate either flow or pressure, when the other quantity is known. The computer model for the directional control valve would be:

Valve in shifted position:

$$Q_A = k_a(P_S - P_A)^{\frac{1}{2}} \tag{13a}$$

$$Q_B = k_b \, (P_B - P_T)^{\frac{1}{2}} \tag{13b}$$

$$Q_N = 0 \tag{13c}$$

Valve in neutral position:

$$Q_A = 0 \tag{13d}$$

$$Q_B = 0 \tag{13e}$$

$$Q_N = k_n (P_S - P_T)^{\frac{1}{2}} \tag{13f}$$

where

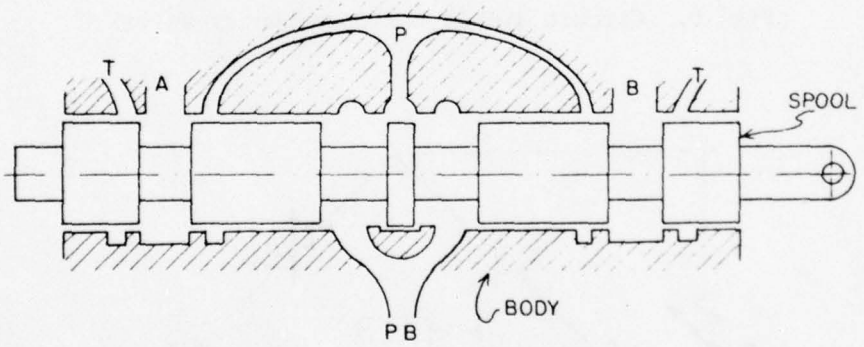
Q_N = Open-Center flow

It is important to emphasize that with the information given in Fig. 4, it is *not* possible to estimate the metering characteristics of the valve.

From the above discussion, it is evident that the actual mathematical manipulations in developing a computer model are no different from those needed for hand calculations. However, they are collected and presented in a general and unambiguous format, applicable to *all* valves of a particular type. It is of interest to note that if the various coefficients of the mathematical models had been presented, the graphs shown in Fig. 1 through 4 could have been generated. Thus, as far as information content is concerned, the mathematical model and the graphs are equivalent. The latter are more compact, while the former are easier to comprehend.

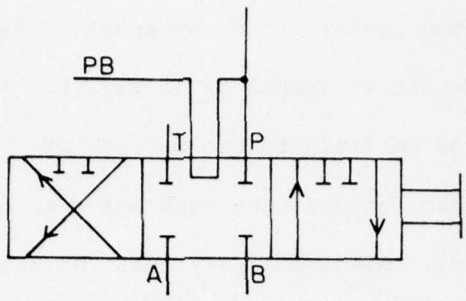
When a hydraulic system consists of a number of components, each of which may be represented by mathematical models, either in graphical or equation format, the problem of ascertaining how the system responds to various inputs can become too tedious for hand calculations. However, using vector notation it is possible to develop compact computer representations which can be adjoined to other component models to obtain system models. This is best illustrated by an example.

Consider an open-center, four-way spool valve with lands as shown in Fig. 5 and circuit symbol as in Fig. 6. As the spool is moved from the center position to one extreme or the other, the lands will connect the pressure and tank ports to one side or the other of the actuator. Simultaneously, part of the flow will be diverted to the next valve, if any, in the circuit, depending upon how far the spool has been moved. The above verbal description can be graphically depicted in Fig. 7 and 8. Figure 7 presents the wheatstone bridge analogy for the four-way, open-center valve, and Fig. 8 shows how various metering orifice areas change with spool position. Note that this model is more complete and realistic than the one presented earlier wherein the valve was either "shifted" or in neutral.



T	TANK
A, B	ACTUATOR
P	PRESSURE
PB	PRESSURE BEYOND

Fig. 5. Cut-Away of an Open-Center Four-Way Valve Showing Metering Lands.



P PRESSURE PORT

PB PRESSURE BEYOND PORT

T TANK PORT

A, B } WORK PORTS

Fig. 6. Circuit Symbol for Open-Center Valve

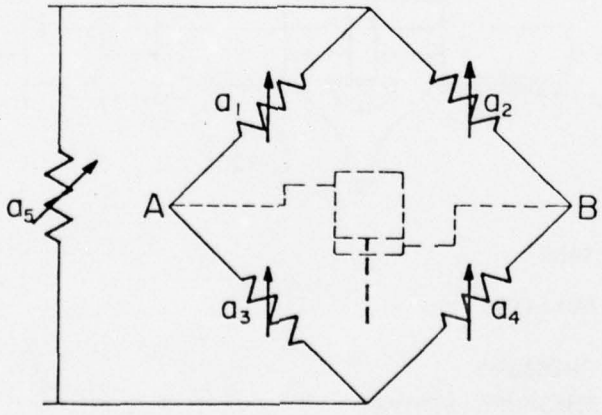


Fig. 7. Wheatstone Bridge Analogy for Open-Center Valve

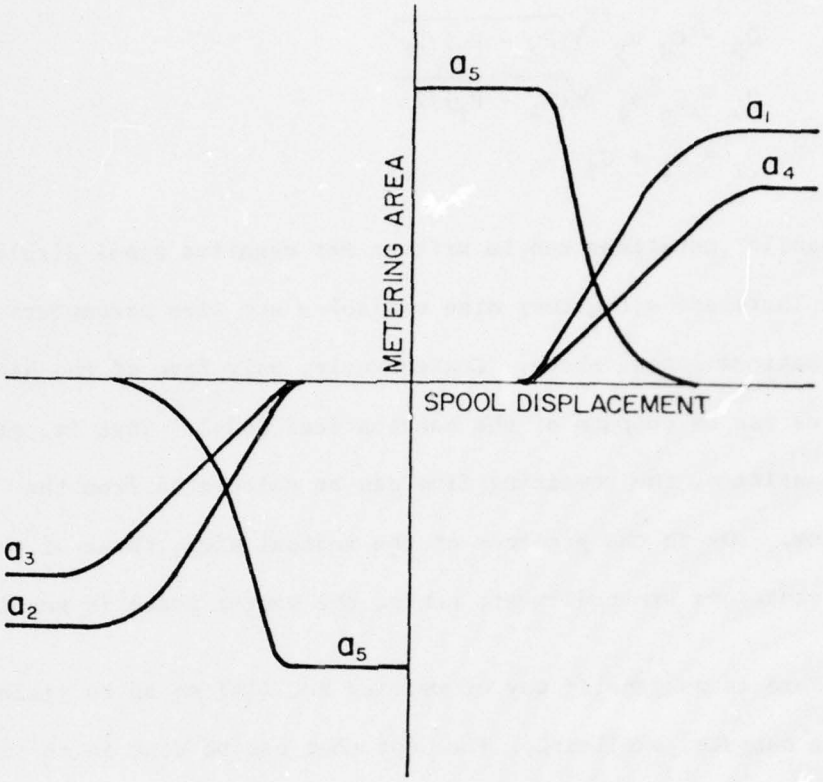


Fig. 8. Change of Metering Orifice Areas with Spool Position for an Open-Center Valve

The mathematical model for the valve is obtained by aggregating the relationships between the various port pressures and flows and the spool displacement. Using the continuity law and the square law for orifice flow and the notation in Fig. 7, the following equations can be written for a positive spool displacement:

$$Q_S = Q_1 + Q_5 \tag{14a}$$

$$Q_1 = C_d\, a_1 \sqrt{(P_S - P_1)/2\rho} \tag{14b}$$

$$Q_5 = C_d\, a_5 \sqrt{(P_S - P_T)/2\rho} \tag{14c}$$

$$Q_4 = C_d\, a_4 \sqrt{(P_4 - P_T)/2\rho} \tag{14d}$$

$$Q_T = Q_4 + Q_5 \tag{14e}$$

Similar equations can be written for negative spool displacements. There are altogether nine variables and five parameters in the five equations listed above. Consequently, only five of the nine variables can be outputs of the mathematical model. That is, given four quantities, the remaining five can be calculated from the equations. Due to the presence of the radical sign, three of the scalar equations are nonlinear; hence, the vector model is nonlinear.

There is no generic way of writing Eq. (14) so as to yield all possible outputs, explicitly. The best that can be done is to express it functionally:

$$\underline{O} = \underline{f}\,(\underline{U}) \tag{15}$$

where the output vector $\underline{O}$ and the input vector $\underline{U}$ may be chosen with some degree of flexibility. When the valve is used in conjunction with a fixed displacement pump, a logical input vector would be $(Q_s\ P_1\ P_T\ Q_4)$ and the corresponding output vector would be $(Q_1\ Q_5\ P_5\ P_4\ Q_T)$. Note that the assignment of inputs and outputs is merely one of convenience and is flexible subject to mathematical requirements being satisfied.

In order to exemplify the synthesis of a system model from the above component models, we give below the static model for a fixed-

displacement pump rotated at fixed speed:

$$Q_p = Q_{th} - k_p P_p \tag{16}$$

where:

Q_p = the actual flow rate of the pump

Q_{th} = the theoretical flow rate of the pump

P_p = the pump delivery pressure

k_p = the pump slip coefficient

Equation (16) assumes that the pump slip flow increases linearly with pressure — any other relationship could have been used.

We shall now assemble all the equations needed to describe an open-center system consisting of a fixed-displacement pump, a four-way valve, and a hydraulic cylinder. For simplicity, we ignore the pressure loss in the lines:

$$Q_p = Q_{th} - k_p P_p \tag{16}$$

$$Q_S = Q_1 + Q_5 \tag{14a}$$

$$Q_1 = C_d a_1 \sqrt{(P_S - P_1)/2\rho} \tag{14b}$$

$$Q_5 = C_d a_5 \sqrt{(P_S - P_1)/2\rho} \tag{14c}$$

$$Q_4 = C_d a_4 \sqrt{(P_4 - P_T)/2\rho} \tag{14d}$$

$$Q_T = Q_4 + Q_5 \tag{14e}$$

$$v = Q_A/A_A \tag{17a}$$

$$Q_B = Q_A - A_B/A_A \tag{17b}$$

$$P_A = P_B(A_B/A_A) + W/A_A \tag{17c}$$

Equation (17) is obtained by considering a cylinder with pressures P_A and P_B acting on areas A_A and A_B respectively; the flow rates through the cylinder ports is Q_A and Q_B and the velocity is v; the external cylinder load is W.

The above nine equations, involving 17 variables, do not reflect the interconnections between the components. Since the pump delivery port is connected to the pressure port of the four-way valve, it is seen that:

$$Q_p = Q_S \tag{18a}$$

$$P_p = P_S \tag{18b}$$

Using similar arguments, the following equations can be written:

$$P_1 = P_A \tag{18c}$$

$$Q_1 = Q_A \tag{18d}$$

$$P_4 = P_B \tag{18e}$$

$$Q_4 = Q_B \tag{18f}$$

Note that corresponding to each pair of interconnected energy ports there are two equations relating the relevant through and across variables.

Equations (14) through (18) thus constitute the static model for the open-center system. Note that, in this formulation, it is easy to identify not only the set of equations pertinent to any component but also to establish the interconnection between the various power ports.

Since all of the six equations used to describe the interconnections between the individual components are very simple and involve only two variables each, we can eliminate six of them by mere substitution to obtain the following set of implicit equations:

$$0 = -Q_S + Q_{th} - k_p P_S \tag{19a}$$

$$0 = -Q_S + Q_A + Q_5 \tag{19b}$$

$$0 = -Q_A + C_d\, a_1 \sqrt{(P_S - P_A)/2\rho} \tag{19c}$$

$$0 = -Q_5 + C_d\, a_5 \sqrt{(P_S - P_T)/2\rho} \tag{19d}$$

$$0 = -Q_B + C_d\, a_4 \sqrt{(P_B - P_T)/2\rho} \tag{19e}$$

$$0 = -Q_T + Q_B + Q_5 \tag{19f}$$

$$0 = -v + (Q_A/A_A) \tag{19g}$$

$$0 = -Q_B + Q_A\,(A_A/A_B) \tag{19h}$$

$$0 = -P_A + P_B(A_B/A_A) + W/A_A \tag{19j}$$

Equation (19) is the same static model for the open-center system, but now the equations constituting individual component models are not identifiable. Since there are nine equations in the mathematical model, the output vector contains nine elements. The input vector is comprised of the remaining three variables. Let us assign elements to the input and output vectors as follows:

$$\underline{O} \triangleq [Q_S \; P_S \; Q_A \; P_A \; Q_5 \; Q_B \; P_E \; v \; Q_T]$$

$$\underline{O} \triangleq [Q_{th} \; P_T \; W]$$

Thus, for a given spool position and a given value of the input vector, the equations can be used to establish the corresponding value of the output vector. In order to keep the mathematical model simple, the metering characteristics of the four-way valve have not been included in it. However, it can easily be appended to Eq. (14) and (19) as follows:

$$a_1 = f_1(x) \tag{20a}$$

$$a_2 = f_2(x) \tag{20b}$$

$$a_3 = f_3(x) \tag{20c}$$

$$a_4 = f_4(x) \tag{20d}$$

$$a_5 = f_5(x) \tag{20e}$$

where x is the spool displacement and f_1 through f_5 are mathematical functions describing the curves in Fig. 8. Consequently, we will rewrite the inputs as follows:

$$\underline{U} = [Q_{th} \; P_T \; W \; x]$$

We could augment the output vector by adding the metering areas a_1 through a_5, but there is generally little need for it.

Let us examine the static model for the open-center system as given by Eq. (14) through (19). We see that, first, the equations by themselves do not indicate which variables are inputs and which are outputs. Second, the set of equations would be slightly different if the spool displacement (one of the inputs) is negative than when it is positive. Third, there are certain intermediate quantities, like orifice areas a_1 through a_5, which have to be calculated whether they are included in the output or not. Last, time is not one of the variables in the input, which implies that the value of the output vector at any time is dependent only upon the input vector at that time.

Typically, a system designer would be interested in how the outputs changed as the inputs were changed. Thus, in an operational cycle, the spool might be moved back and forth, and the actuator load would also be changing, and the system designer would like to know how the pump delivery pressure and actuator pressures change in the course of the cycle. In order to use the above model to develop the required times histories, all that is necessary is to repetitively solve the

set of equations representing the system for a set of input vectors which adequately describes the input time histories. Note that a complete solution of the set of equations is needed, even though the system designer may be interested in only one or two quantities in the output vector. This is because the equations are "coupled" (i.e., some variables enter into two or more equations).

The process of generating the output time histories by repetitively solving the mathematical model for a given input trajectory is called simulation. If the mathematical model is realistic, the values of the simulated output should agree with that obtained by subjecting the actual physical system to the same input time history. We will now give an example of such simulation done on a digital computer. The role of the computer is to solve the set of equations (14) through (20) repetitively for the prescribed input time history.

The system being simulated is similar to the open-center system discussed above, except that it uses two identical cylinders in parallel fed from the same four-way valve. The cylinders are assumed to share the flow as well as the external load equally. The actual program used is written in FORTRAN, but an equivalent program can be written in any high level scientific-oriented language.

The four-way, open-center valve has metering characteristics as shown in Fig. 9. We mentioned earlier that the mathematical model for the valve could include equations describing the metering areas as

functions of the spool position. In this instance, the model will be stored numerically (i.e., as sets of area vs. spool displacement for each metering orifice). Such data are easily acquired by subjecting a valve to a metering characteristics test. Using the spool displacement time history furnished in the input, the program calculates the metering orifice areas for each point in time, interpolating between data points if necessary.

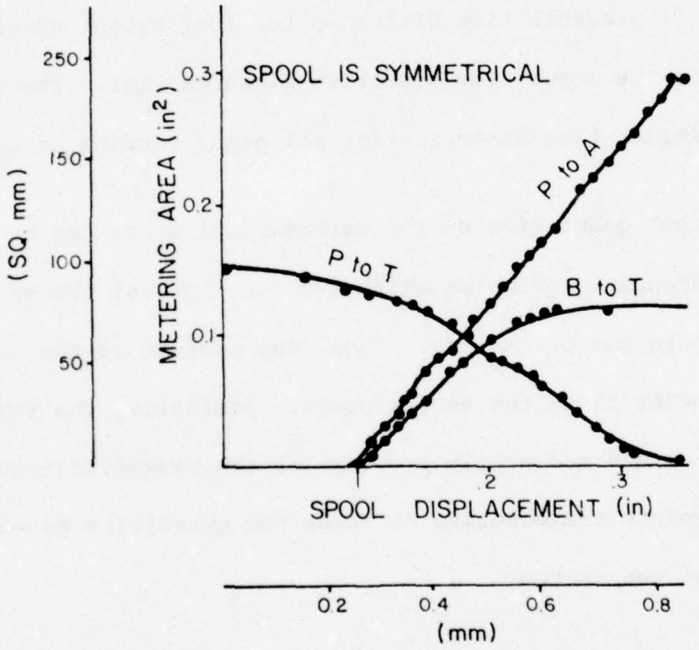


Fig. 9. Metering Characteristics of Four-Way Valve, Obtained Experimentally.

The mathematical model for the actuators is input by reading in the cylinder bore and rod sizes. The pump model, as given in Eq. (16),

is furnished to the computer by reading in the theoretical flow rate and the slip flow coefficient.

Figure 10 presents input time histories for a typical simulation. The pump theoretical flow rate Q_{th} and the tank port pressure P_T are constant for the duration of the cycle and consequently not depicted in the figure. For convenience, the abscissa is normalized time, the time for the entire cycle being unity.

Figure 11 presents time histories for four output quantities corresponding to the input time histories given earlier. The computer generates similar time histories for all other outputs as well.

The output quantities of the mathematical model can be used to calculate secondary variables which are functions of the outputs and possibly inputs and parameters. Thus, the product of the actuator load and the velocity gives the output power. Similarly, the product of the pump flow rate and supply pressure is the hydraulic input power. Figure 12 depicts trajectories of these two quantities as well as the efficiency of the system.

In order to examine the behavior of the system to a different input time history, all that is necessary is to change the cards from which this information is read before running the program. If the designer is interested in examining the effect of changing a component, say the pump, this is easily accomplished by specifying the new flow

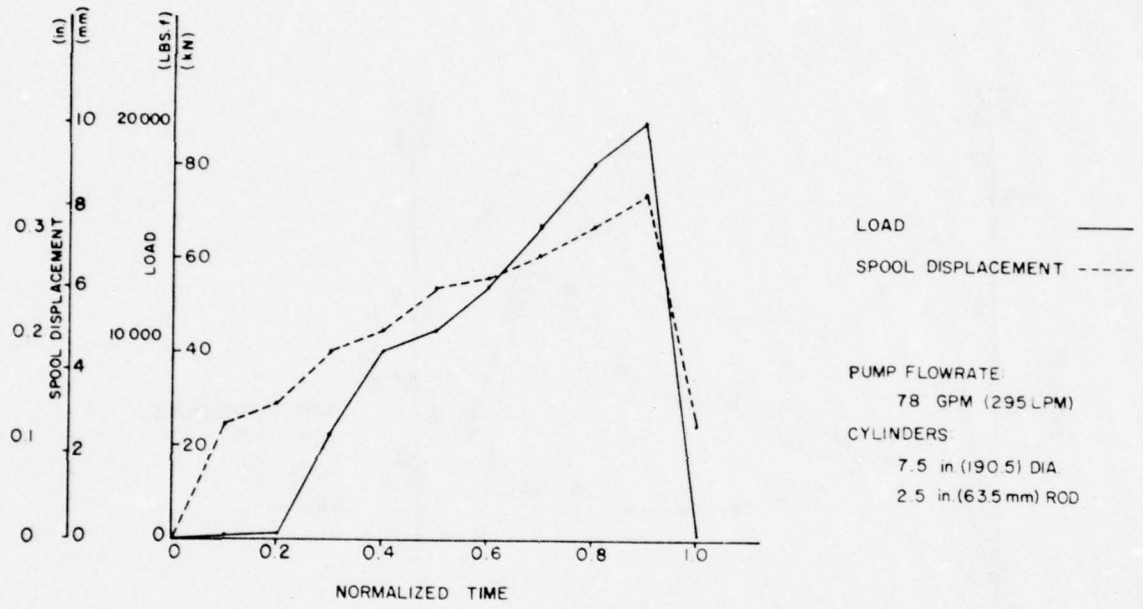


Fig. 10. Input Time Histories For Open-Center System

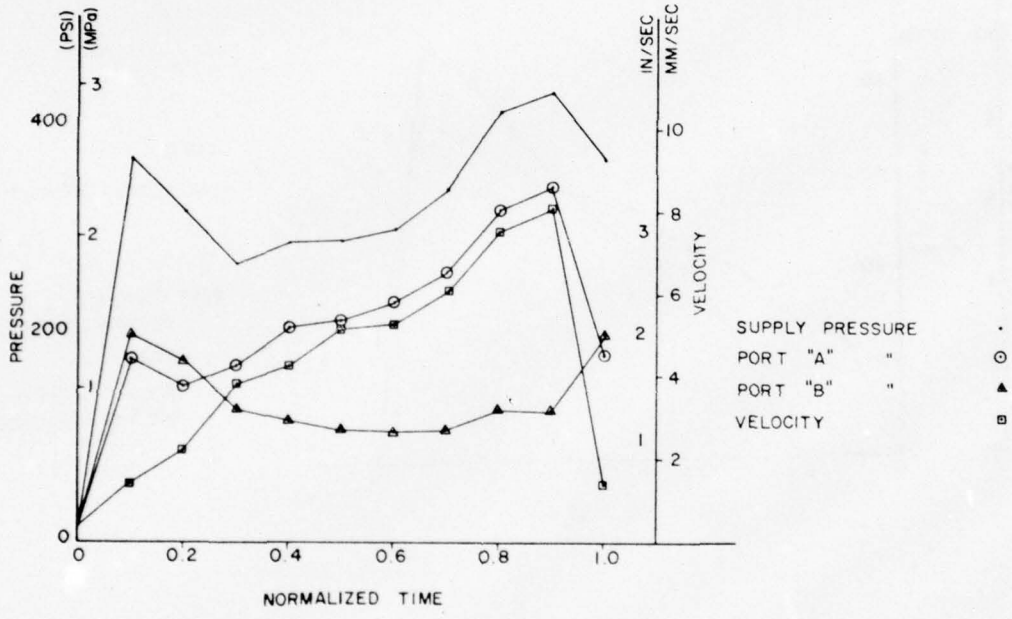


Fig. 11. Output Time Histories For Open-Center System

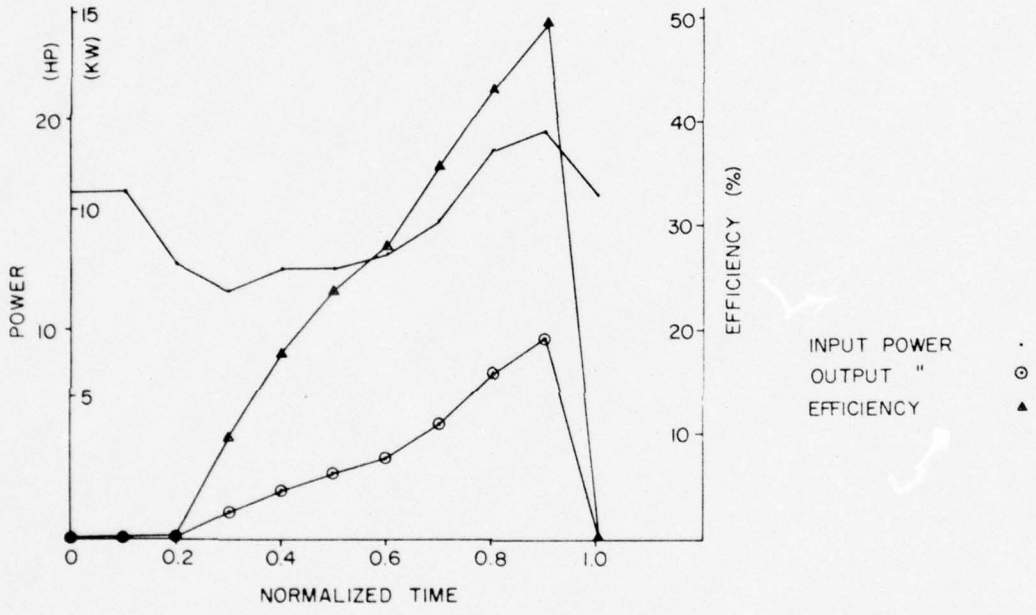


Fig. 12. Time Histories For Secondary Quantities For Open-Center System

rate and slip coefficient in the data cards fed to the computer. Obviously, the effort needed to effect such changes in the model is much less than that needed to actually change a piece of hardware on the physical system.

CHAPTER III

REVIEW OF COMPUTER SOFTWARE

The main objective of mathematical analysis of hydraulic systems is to minimize the trial-and-error stage of product development. Additionally it can be used by equipment users to appraise alternative designs of machines and to project the performance of a degraded machine. As systems become more and more sophisticated, it becomes extremely tedious and time-consuming to perform such analysis by hand, even after making simplifying assumptions. Additionally, the simplifying assumption often result in unacceptably large margins of error and non-optimal "conservative" designs. However, the development of the general purpose computer in the last two decades has provided the analyst with basic tools to appraise a variety of physical systems; e.g., structural assemblies, heat exchangers, electrical networks, mechanical linkages, etc. The computer programs in many of these areas have been refined to such a state that the average program user need not know, other than in a general sense, about either the mathematical relationships underlying the analysis or the algorithmus used to obtain the computer solutions. Examples of such programs are NASTRAN for structural analysis and ECAP for electrical circuit analysis. In the area of hydraulics, the development and use of computer software has not become as widespread as in structural or electrical circuit design. However, significant progress has been made in the past five years, and as more and more designers and managers are apprised of the cost-

saving potential of computerized analysis, software development will accelerate. In the remainder of this chapter we will describe in general terms the scope of activities delegated to the computer in engineering and scientific analysis, discuss some of the mathematical and non-mathematical processes underlying all such analysis and review computer software currently available for hydraulic systems analysis. Figure 13 presents the computer analysis pyramid, which shows the hierarchial relationship between various tasks delagated to the computer. Each level in the pyramid relies on the lower levels, which are often "transparent" to the user. It is advantageous to refer to the pyramid while reading the next two sections.

THE COMPUTER IN ENGINEERING ANALYSIS

Too many engineers still perceive the digital computer as an electronic version of the old desk calculator. It is true, however, that the fundamental operations performed by a computer are arithmatic in nature. Even the analysis of Boolean expressions, needed to make branching decisions, is based on arithmetic. The average engineering or scientific user of the computer is introduced to it through a high-level language such as FORTRAN or PL/I. He is taught to convert mathematical expressions into equations and, using the logic commands available, to calculate the numerical values of whatever quantities he is seeking. A number of engineering users of computers find this level of competence sufficient for a majority of design/analysis problems.

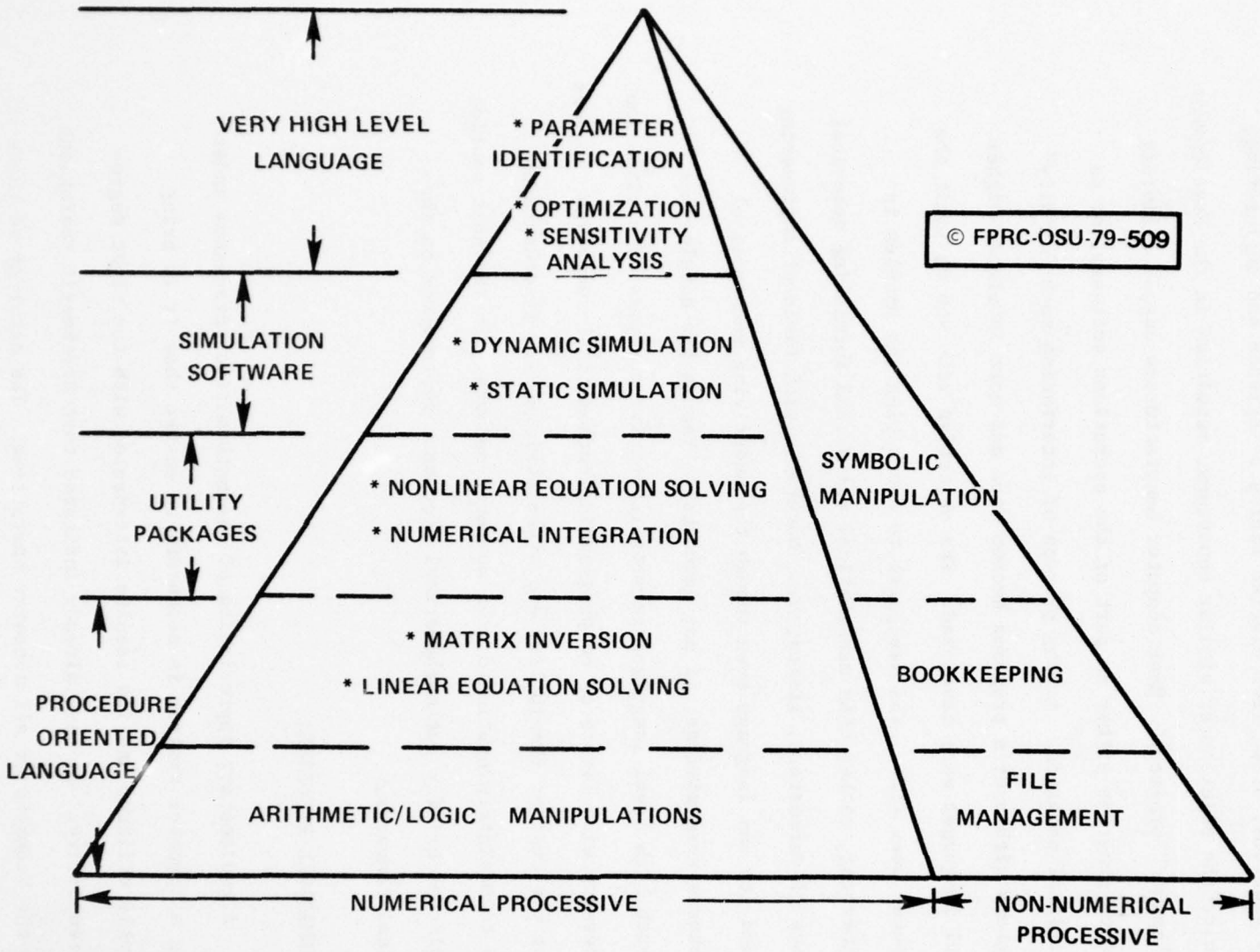


Figure 13. The Computer Analysis Pyramid

Many problems, drawn from diverse fields of science and engineering, require the solution of similar equations, resulting in the development of "canned" programs. Most computer manufacturers supply a library of such programs either as part of the operations software, or as additional packages. As the process of interfacing such "utility" programs with user's programs became more and more involved, higher level languages were developed. The objective here was to limit the computer user (i.e., the designer) to describing the problem in engineering, rather than mathematical terms, and furnishing numerical values of parameters, limits, etc. Naturally each field of engineering needed its own language even though the underlying mathematical processes were similar, if not identical. Though the development of such high level languages permits repetitive analysis of a situation by persons with little or no programming/mathematical knowledge, it does so at the cost of flexibility and often efficiency. It could also lead to misinterpretation of the computer outputs. In the next section we discuss briefly the mathematical processes implemented on the digital computer.

MATHEMATICAL PROCESSES

A rudimentary appreciation of the mathematical processes underlying a computer program is necessary to ensure that it is being properly utilized and its results interpreted with the right degree of credibility. We have already indicated that arithmetic operations form the backbone of all computer operations. The accuracy of these

operations depends on the "word size" of the computer and the output format. For example the IBM 360 and 370 computers have a word size of 4 bytes, and using exponential notation, could furnish numbers with a maximum of 7 significant places after the decimal point. Hence in a number given as 8.4434865 the last one digit, and possibly the last four are probably unreliable. The danger lies in subtracting two such numbers and ascribing significance to the difference especially when it is small. The computer manufacturer can usually furnish the accuracy of the arithmetic operation in a specific machine. Many computers are capable of extended precision--the IBM 360 and 370 machines referred to can have words of 8 bytes in "double precision" operation. Accuracy is thereby increased, but is accompanied by increased computational costs. Another aspect of computers needing attention is output accuracy. Output formats are tailored to suit the majority of users. The number of significant digits printed out is almost always less than that carried along in the calculations. The last digit is often the result ot truncation rather than rounding off. Thus 3.3496855 would be printed out in F10.2 Format as 3.34 and not 3.35. Once again the difference between two almost equal numbers may not reflect reality.

The calculation of the numerical values of expressions accounts for a significant portion of many engineering and scientific computer programs. Logical decisions, using FORTRAN statements like "GO TO" and "IF" and "DO", account for much of the remainder. At a higher level

of analysis, though, large segments of programs can be identified with equation solving. In a very trivial sense each FORTRAN equation "solves" for the left hand side. However much of engineering analysis involves the solving of single or multiple equations. The solution of a set of linear equations was one of the first tasks attempted on the digital computer and currently almost all computer installations include one or more linear equation solving subprograms in their library. Such subprograms may be called iteratively by the main program and quite often the user of a "canned" program is unaware of the linear equation solving entailed by his computer usage. The "failure" of a software package to "deliver the goods" has been traced, on more than one occasion, to poor accuracy or even failure of the linear equation solving subroutine. Even though the average "canned" program user cannot be expected to diagnose the causes of failure of such subroutines, he should nevertheless be aware, in general terms, of the circumstances under which failure can occur. For example, if a set of linear equations is ill-conditioned, the accuracy of the solution is suspect, unless algorithms specifically written for ill-conditioned equations, are used. A large part of structural analysis programs, e.g., NASTRAN, and vibration analysis program consists of linear equation solving. Hence their users have to confirm that no ill-conditioned equations are being processed.

Somewhat similar to linear equation solving, and also offered in most computer libraries, is matrix inversion. The accuracy of

matrix inversion depends on the "condition" of the matrix and the algorithm used. Brute force approaches to matrix inversion become very inefficient and possibly inaccurate once matrices exceed a dimension of ten by ten. The solution of partial differential equations by finite element and finite difference techniques gives rise to large and usually "sparse" matrices. It is only recently that general subroutines for handling large sparse matrices efficiently and quickly have been introduced. Software packages which utilize these subroutines can be expected to be more efficient than older ones.

The solution of nonlinear equations has been one of the first tasks assigned to the digital computer. Nonlinear equations in one variable are generally solved on the computer by a trial and error process. It is often referred to as "root finding". Many mathematical techniques are available for this, none of which is universally applicable. If one technique fails, others should be tried. Nonlinear equations in multiple variables are most conveniently treated as vector versions of the scalar nonlinear equation. The complexity of the problem is in no way reduced, though notation is simplified. Since nonlinear equations have, in general, multiple solutions, there is usually no guarantee that the computer will, of itself, reach the physically meaningful solutions. However by "boxing in" the variables, and furnishing initial trial values for them, chances of obtaining an accurate and quick solution are greatly increased. A number of nonlinear equation solving programs use matrix inversion and linear

equation solutions as intermediate steps. Hence the accuracy of a solution depends firstly on the nonlinear solution algorithm, secondly on the utility subroutines it accesses and thirdly on the accuracy of the arithmetrical operations.

In the simulation of dynamic physical systems, of which hydraulic systems are an example, a common requirement is the numerical solution of scalar or vector nonlinear differential equations. Such solutions are generated by a process known as numerical integration, for which many algorithms have been developed. Some of these are explicit integration methods and need only iterative evaluation of numerical expressions, while others are implicit integration methods and require the solution of a set of nonlinear equations. Since numerical integration advances the solution in discrete steps, and each step involves iterative calculation of expression or solution of nonlinear equations, the number of arithmetical steps involved in generating a numerical solution is truly enormous. As the number of arithmetical operations increases, the effect of finite word length is to reduce the accuracy of the final answer. Thus, in the final stages of numerical integration of a set of equations, the number of significant digits may be only two or three, even if seven digits are written out. One method of countervailing this tendency, at a price, of course, is to use double precision numbers.

All the discussion above has centered on basic mathematical operations and is valid for all scientific and engineering analysis.

In order to use any of the utility subprograms or algorithms to simulate a system, it is necessary to write the mathematical model in a form compatible with the utility subprograms. This generally requires the writing of a main program, auxiliary subroutines and input-output formats—operations which require a good working knowledge of FORTRAN and sufficient mathematical expertise to select and interface the correct utility algorithms. With a view to making digital simulation a viable tool, systems analysts have developed general purpose simulation packages wherein the user inputs the mathematical model and control parameters in a prescribed format. A preprocessor, in effect, uses this information to write the main program and other modules required to interface with the utility subprograms. Almost all of these packages appear to be written for electrical networks or systems which can be modeled using electrical analogies. With few exceptions, all such software is intended for dynamic simulation. Some can be used, although inefficiently, for static simulation. It can safely be said that few are suitable for static analysis of hydraulic systems.

Even though digital simulation can be used merely to examine the dynamic or static behavior of a system, the designer or analyst will often consider this as only an intermediate step. He is often interested in establishing the sensitivity of the system behavior to changes in design parameters, or in selecting the "best" parameters for a prescribed system, or in estimating the numerical value of one or more parameters inaccessible to measurement. All of the above kinds of

analysis can be performed by using the results of iterative digital simulation. It should be pointed out that many simulation packages are not geared for parameter identification, optimization or sensitivity analysis and often have too rigid a structure to be modified for such analyses.

CURRENTLY AVAILABLE SOFTWARE

This review is not exhaustive but it will discuss the salient features of all the major varieties of computer programs usable for hydraulic system simulation. The word "usable" implies that quite often the programs have been written for other systems and some effort may be needed to translate the problem to a form suitable for the program.

Static Analysis

The objective of static analysis of a hydraulic system is to calculate the pressures and flows in different parts of a circuit as the machine performs a well defined task. Auxiliary quantities like cylinder velocities, input and output power and efficiency may also be calculated. It should be noted that static analysis does *not* mean that actuators are stationary or there is no flow; it only implies that dynamic effects due to accelerating masses, fluid inertia or compressibility are absent. In mathematical terms, the system model is a set of (nonlinear) algebraic functions, which have to be solved iteratively, as the inputs change.

FPSA

*F*luid *P*ower *S*ystems *A*nalysis is a FORTRAN program developed by Hsu and Hammerle [Ref. 1] for analyzing flows in networks comprised of lines, valves and actuators. Motor torques and actuator loads have to be constant. Only one cylinder can be present and valves have to be modeled as orifices of a known and constant area. The program calculates the pressure drop in all lines and performs a flow-balance iteractively until consistent results are obtained.

Because of the limitation on the number of actuators, the kind of valves that are included, and the inability to handle varying loads or human inputs, this package will find very little use in the mobile hydraulics field.

SSFAN

*S*teady *S*tate *F*low *AN*alysis is one of the programs developed by McDonnell Douglas for analyzing aircraft hydraulic Systems [Ref. 2]. It is modular in nature and has a library of models for components such as, check valves, accumulators, heat exchangers, tees, elbows, and fittings in addition to variable displacement pumps and servoactuators. The user describes the configuration of the system and furnishes numerical values of all parameters relevant to the components in the circuit. The program calculates flows and pressures for constant external loads. No manually operated valves are currently included in the library. Without significant modifications, the program is likely to be of limited

usage for industrial and mobile hydraulic systems, especially those requiring human inputs.

OCSYS

*O*pen *C*enter *SYS*tem analysis program has been developed at the Fluid Power Research Center, Oklahoma State University, for analyzing single actuator open-center mobile hydraulic systems [Ref. 3]. It has been extended for thermal analysis and operational severity analysis [Ref. 4, 5].

The program solves a set of nonlinear equations for establishing the pressures, flows and actuator velocities for a prescribed valve position and cylinder load on a given system. By repetitively solving these equations, it establishes the time histories of the various pressures and flows for a given operational duty cycle. These quantities, in turn, can be used to calculate both instantaneous and average power input and output and other dependent quantities.

In its present form the program is limited to single actuator systems, but could be expanded to multiple actuator systems. However it is not usable for analyzing other configurations, e.g., double pump systems, load sensitive closed-center systems, etc. The user is expected to know FORTRAN.

Dynamic Analysis

The objective of dynamic analysis of hydraulic systems depends on

whether the system is a control system or a power system. Examples of control systems are machine-tool and aerospace servo-systems and hydrostatic transmission; examples of power systems are hydraulic presses, and implement drives in mobile equipment. In the case of control systems, dynamic analysis is performed to:

(a) examine frequency response

(b) ascertain stability

(c) investigate transients.

Most control systems can be treated as single-input single-output systems. More complex systems can be analyzed by superposing the simulation of constituent single-input single-output systems. Almost all control systems can be approximated by linear dynamic models for preliminary analysis, which can be done in the frequency domain. Melsa and Jones have developed a very useful package of digital computer programs for frequency response analysis [Ref. 6]. The programs are written in FORTRAN and can be used either independently or in conjunction with main programs written specifically to analyze certain kinds of systems. In the case of power systems, especially those using a human operator, dynamic analysis is performed primarily to investigate transients. In some cases, it may be necessary to examine the stability of selected components, e.g., valves.

Programs for dynamic simulation in the time domain can be classified as either general network or systems analysis programs or hydraulic system simulation packages. Some use languages of their own,

and hence cannot be easily modified by the user; others use a preprocessor which can be bypassed by the knowledgeable user. We describe below some software in this area.

SUPER-SCEPTRE

SCEPTRE is a program representative of the generalized network approach. It is, in the words of its developers, "an automatic circuit analysis program capable of determining initial conditions, transient and steady state responses of large networks" [Ref. 7]. SUPER-SCEPTRE is a preprossessor, developed for use with SCEPTRE, aimed at simulating multi-degree of freedom mechanical systems (8). Generalized network parameters, e.g., inductance, capacitance and resistance, are used to describe the system. Before using the program it is advisable, if not necessary, to draw the network analog for the system under consideration. Since the input information has to be furnished in the form required by the preprocessor, it is difficult to gain access to the numerical processing subroutines.

The package is, in its present form, unsuitable for parameter identification or optimization. Nor is it suitable for static analysis in the sense OCSYS is. Additionally, SCEPTRE uses Runge-Kutta and trapezoidal rule algorithms for numerical integration, which makes it not only inefficient at times but also fails under certain circumstances.

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MARSYAS

This program was developed at the Marshall Space Flight Center for analyzing large aerospace systems, primarily in the frequency domain [Ref. 9]. All nonlinear components have to be dissected down to the level of nonlinear (and linear) elements described by *scalar* nonlinear/linear equations. Though MARSYAS is intended for "large" systems, it has no features specifically recommending it for hydraulic system simulation.

IMP

This is described as a "Software System for the Direct or Iterative Solution of Large Differential and/or Algebraic Systems" [Ref. 10].

It differs from the preceding two packages in being a utility program; i.e., in order to use it, the user has to write a main program and auxiliary subprograms to handle graphical displays, input and output. A good knowledge of FORTRAN is obviously necessary for performing the interfacing. It has an efficient numerical integration algorithm which can handle large and stiff systems very efficiently.

No information as to the actual usage of IMP in hydraulic system simulation has been published. For an analyst skilled in mathematical techniques, however, it has all the prerequisites for development into a viable software package for both dynamic and static simulation.

HYTRAN

This is another of the programs developed by McDonnell Douglas Corporation for analyzing aircraft hydraulic control systems [Ref. 11]. Like SSFAN, it has a library of models and the user furnishes numerical values of various parameters and describes the system configuration. HYTRAN models all interconnecting piping by the method of characteristics, and dynamic components, such as variable displacement pumps and servo-actuators, by lumped parameter differential equations. Additionally, it uses a fixed step size for numerical integration.

In principle, HYTRAN is modular, so that by adding appropriate models to its library, almost any system can be analyzed. In practice, the processing of adding a library member requires considerable knowledge of the innards of HYTRAN as well as numerical analysis techniques. The net result is that the program is rather inefficient and too rigid to be usable for a large class of mobile hydraulic systems.

HYDSIM II

This program was written to simulate complex hydraulic systems using multi-port component models [Ref. 12]. It has many features in common with HYTRAN, but allows components to be connected without intermediate lines. However certain types of connections are "forbidden." A library of component models is maintained and can be accreted. The program relies on Runge-Kutten and Adams-Moulton

algorithms for numerical integration. As indicated earlier, these routines may fail in certian cases, a failure not shared by IMP and other routines using implicit integration techniques. HYDSIM II is also not geared for static analysis.

HSFR

Another of the programs developed by McDonnell Douglas Corporation, this program can be used to analyze the propagation of fluid borne noise in a hydraulic system [Ref. 13]. The program libraries includes models for transmission lines (with friction), accumulators, filters, orifices and attenuators, in addition to a piston pump. This program is smaller than HYTRAN or SSFAN and easier to modify to suit mobile hydraulic applications. Like the others it is primarily a simulation program and will require substantial modifications for optimization.

CSMP

*C*ontinuous *S*ystem *M*odeling *P*rogram was one of the first general purpose dynamic system simulation programs made available to IBM computer users [Ref. 14]. CSMP uses a preprocessor to convert user input into FORTRAN statements. One of the conveniences offered by CSMP (and similar packages) is the flexibility of coding and input/output formats. The numerical integration algorithms in this package are not suitable for stiff systems but can be modified by the knowledeable user. Error diagnostic messages are often confusing, if not misleading.

Use of a preprocessor implies that large core storage areas are needed even for analyzing small systems. Because of its wide availability, CSMP is well known to analysts in the fluid power industry. It is not exactly suited for either static analysis or optimization.

ACSL

This software package resembles CSMP in its general structure, but is claimed to be more economical because of efficient "book-keeping." It contains an implicit integration algorithm which makes simulation of stiff systems accurate and efficient, and gives better diagnostic messages. Currently ACSL is available fo IBM 360/370, CDC 6000/7000 and UNIVAC 1100 machines [Ref. 15]. Like CSMP it is not suitable for either static analysis or optimization.

ECAP II

*E*lectronic *C*ircuit *A*nalysis *P*rogram II is the second generation version of a package developed by IBM. It is so radically different in its internal structure that the retention of the acronym is likely to mislead some potential users. It is one of the few languages which is truly hierarchic in structure and geared for analyzing large systems, albeit in network terminology [Ref. 16]. Using as it does, both implicit integration and sparse matrix handling techniques, it is very efficient at both steady state and dynamic analysis. Its

elaborate bookkeeping allows the user considerable flexibility in developing and storing subsystem models, at the same time ensures efficient use of computer storage space through dynamic allocation. ECAP II is suited for neither static analysis of hydraulic systems nor optimization without substantial changes to its precompiler. Since it is the property of IBM, users may not be permitted access to its coding and modify it for their use.

DYSIMP

The *D*ynamic *SIM*ulation *P*rogram has been developed by researchers at Oklahoma State University, the most recent version being the work of Ebbsesn [17]. Patterned, to some extent, after CSMP, it offers the user much more flexibility in appending FORTRAN subroutines to perform desired calculations, or over-ride those built into the package. The program can be modified fairly easily for parameter identification and optimization. Input and output is free format and the user can generate a variety of plots either on a high speed printer or on an off-line plotter. A variety of utility subroutines permit the calculation of cumulative information, solution of linear and nonlinear equation sets and interpolation. A version for optimization is available.

HYDS

This program was specifically developed for the analysis of open-center hydraulic systems [Ref. 18]. Its current version handles two

actuators and implements an integration routine very similar to that in ECAP II. Written in FORTRAN and using no precompiler/preprocessor, it requires some knowledge of FORTRAN on the part of the user. Mathematical models of the components are furnished in subroutines and changes in the order of the model, in the course of simulation, can be accommodated by switching models, each of which is stored in a separate subroutine. HYDS can be used for static analysis, and incorporated into an optimization/parameter identification package.

PROSE

The authors of this package claim that it is a "general-purpose scientific programming language with syntax and semantics for addressing problems at the calculus level as well as the algebra level" [Ref. 19]. The language is hierarchical in structure and includes a compiler, a processor which permits translation to FORTRAN, a library of utility programs, and bookkeeping routines. Its unique feature is the integration of optimizing with simulation and equation solving into one coherent package. Even if only a fraction of its claims are true, the language should take a considerable portion of the "precomputer" analysis burden off the shoulders of the analyst. Currently PROSE is being suported on CDC computers.

Closure

The status of computer simulation packages is in a state of flux. Currently fluid power systems analysts have the choice of either

relying on general purpose simulation languages, like CSMP, or developing their own packages, like HYTRAN. The volume of simulation is not yet large enough to attract software specialists to develop a language oriented towards *all* facets of the fluid power industry. The needs of one arm of the industry, e.g., the aircraft applications, are *perceived* to be quite different from that of the others, e.g., the industrial and mobile equipment applications. Simulation of static and dynamic behavior of components and systems has been restricted to the large end-item and component companies. The main reason for this state of affairs appears to be lack of sophistication on the part of the equipment-buyer/user. It is true that in the case of many hydraulically powered equipment, the performance requirements are such that simple hand calculations are sufficient to establish that they are met. In other cases, however, analysis of operational severity, efficiency, heat generations and response to human inputs would benefit considerably by resorting to simulation. Component manufacturers currently furnish some information on static behavior of off-the-shelf hardware, and would probably furnish more for special designs or customized components. Only after the need to furnish either mathematical models, or sufficient test data to develop (empirical) mathematical models has been accepted by all parties concerned, is it possible to establish standard mathematical formats to communicate the model. In the meantime analysts in different companies and government agencies can be expected to evolve their own software to suit the equipment, systems and components coming under their purview.

CHAPTER IV

MACHINE AND COMPONENT STATE ASSESSMENT

The user of hydraulic equipment is always cognizant of the fact that the failure of even a small component may incapacitate a machine for an extensive time period. Hence, any technique which can be used to ascertain the state of a machine, in so far as its competence to perform a prescribed mission is concerned, without physically dimsantling it, should be of interest to equipment users. Computer models of components and systems can be used to assist in machine state assessment. The remainder of this chapter describes how this can be performed. Even though most of the discussion centers around hydraulic components and machines, the same methodology applies to other mechanical hardware, used in conjunction with or separate from electrical controls.

Figure 14 depicts the overall scheme for machine state assessment and will be used as the basis of this discussion. It is appropriate to describe the contents of each "box" and relate them to the contents of earlier chapters of this report.

MANUFACTURERS' BROCHURES AND TECHNICAL SPECIFICATIONS

Chapter II has indicated how manufacturers of components furnish technical information on their products and how this can be used sometimes to develop mathematical models. Even though the information presented in the brochures, or supplied on request by the vendor, is

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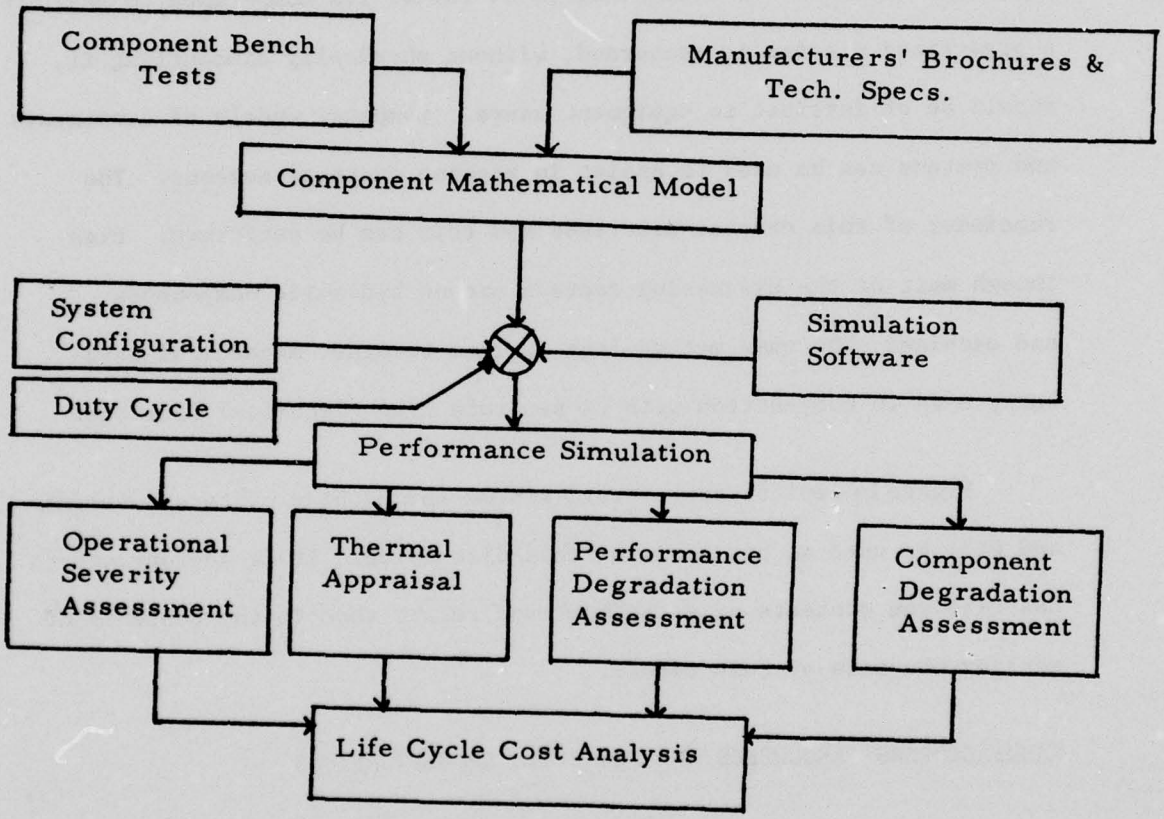


Fig. 14. Flow Chart of Component and System Model Usage for Machine Assessment

often not complete enough to develop a comprehensive mathematical model, it can be used to provide at least some of the parameters in the model. For example, there is no need to physically measure the bore, stroke or rod diameter of a hydraulic cylinder, if the manufacturer's brochure lists them. However, the leakage rate and drag force may need to be explicitly measured. Almost all technical specifications furnished by component manufacturers pertain to performance and little information is provided on contaminant sensitivity or endurance. However, this situation is changing, partly due to procurement specifications calling for specific requirements in these areas, and partly due to standardization activities at the national and international levels.

COMPONENT BENCH TESTS

Apart from verifying manufacturer's claims, the purpose of bench tests is to collect enough data to develop component mathematical models. Chapter II has indicated general requirements for test procedures so that empirical mathematical models can be developed from test data. These tests measure only those variables which are normally accessible to the component user, i.e., the hardware is not modified to obtain additional parameters. As an example, if a relief valve is being tested, no attempt is made to measure the displacement and velocity of the internal poppet or spool, nor is the valve dismantled to measure geometric dimensions or spring rates. Hence, any mathematical model developed from "external" measurements, can not be used to predict

"internal" variables. Another point to be made is that extrapolation from test conditions should be done with caution. Thus, if all tests on a valve have been performed for a temperature range 30°C — 50°C, the mathematical model may be valid up to 60°C if turbulent flow is a reasonable assumption. However, the model may be quite erroneous in predicting behavior at 10°C if the flow has changed from turbulent to laminar. It is important to emphasize that a component mathematical model, developed from bench test data, can be incorporated in the mathematical model for any system using the hardware. Thus, the analyst can simulate the behavior of a system before it is assembled, by combining the mathematical models of the proposed components, each of which might have been tested for entirely different purposes.

COMPONENT MATHEMATICAL MODEL

As explained in earlier chapters, the purpose of a mathematical model is to convey information succinctly and unambiguously. Performance models, whether static or dynamic, interrelate the performance variables of a component, typically flows, pressures, forces and speeds for hydraulic components. All models involve parameters which are generally invariant. Some of these are geometric, e.g., cylinder bore, pump theoretical displacement, etc., while others can only be calculated from test data, e.g., pump leakage flow coefficient, cylinder drag force, etc. A component model should be invariant, i.e., it should not be contingent upon the system in which the hardware is incorporated; however, this is not always the case. Thus, for example most mathe-

matical models for servo-valves assume that the load is an equal area cylinder. A general model for a hydraulic component will be, in almost all cases, multi-input multi-output and nonlinear. In mathematical terminology, if the model is static, it will consist of a set of algebraic equations, and if it is dynamic, it will consist of a set of differential algebraic equations. In both cases there will be constraints on certain variables to reflect physical realities, e.g., pressures cannot fall below zero and cylinder displacements are limited by stops or other mechanical restraints. It should be noted that both static and dynamic models can be used to evolve time histories of output variables.

SYSTEM CONFIGURATION

Strictly speaking, the performance of an individual component cannot be simulated. By assigning numerical values to the inputs to a component model, one implicitly describes a system comprised of the component and its environment. Thus, a pump can be simulated only by (conceptually) attaching a downstream load, which may be a needle valve or an actuator. Similarly a relief valve needs at least a tank connection and a pump upstream.

The manner in which different components of a system are interconnected is called the system configuration. Within certain limitations, the same components can be interconnected in different ways to obtain different circuits. Hence, if a digital computer is to be used to simulate the performance of a system, or an individual component

incorporated in it, there should be an unambiguous way of describing them. Currently, only a few programs, specifically written to analyze hydraulic systems, contain provisions to describe the interconnections between different components. Examples of these are HYDSIM [Ref. 12] and HYTRAN [Ref. 11]. General purpose simulation packages like CSMP [Ref. 14] or ECAP II [Ref. 16], which rely on network theory contain "nodes" and interconnecting "elements" or "subcircuits". A hydraulic component may have multiple "nodes" and their interconnections may be quite different from the actual physical interconnections of the components. In HYDS [Ref. 18] the system configuration is described in a set of FORTRAN equations which are distinct from the equations describing the individual subsystems.

DUTY CYCLE

Almost all mobile hydraulic machinery are engaged in tasks which can be described by duty cycles. Many machines are capable of multiple tasks which may involve one or more subsystems operating sequentially or simultaneously. In some cases it is possible to focus on a subsystem, e.g., the hydrostatic steering on a lift truck, and analyze it in isolation. However, when a common pump feeds many actuators, as in the case with many open-center systems, the operation of one subsystem influences, and is influenced by, other subsystems. In this situation the entire machine has to be analyzed for a specific mode of operation. The typical mobile hydraulic system consists of one or more pumps

feeding one or more cylinders or motors, which are controlled by directional control valves. Relief, counterbalance, flow control and other types of valves are generally present. The movements of the directional control valves, whether manual or via electrical/mechanical signals are inputs to the system. Generally speaking, the mechanical loads/torques on the actuators also have to be treated as inputs to the system. The pressures and flows in various subsystems, and the actuator velocities are the outputs. Hence a duty cycle simulation requires a time-history of all valve motions and corresponding actuator loads. The system configuration and component parameters, e.g., cylinder diameters, pump displacements and relief valve settings are invariant for a specific duty cycle.

SIMULATION SOFTWARE

Chapter III has provided a comprehensive survey of digital computer software for hydraulic system analysis. As indicated therein there are very few software packages suitable for static analysis of mobile hydraulic systems. The general purpose simulation languages, e.g., CSMP, are primarily meant for examining transient behavior of dynamic systems, and would be very inefficient for simulating complete duty cycles. The knowledgeable analyst can usually write his own static analysis program for a specific system by using utility programs for nonlinear equation solving. An example of this, written for a single actuator open-center system is given by Iyengar [Ref. 5].

PERFORMANCE SIMULATION

The objective of performance simulation is to obtain time-histories of the hydraulic variables, i.e., pressures and flows in different parts of a system, as the machine performs a duty cycle. In the case of complex systems, consisting of multiple actuators, pressure and flow control devices and arbitrary operational cycles, it is usually convenient to calculate *all* pressures and flows, even though some of them may not be of immediate interest. Once the hydraulic variables are ascertained, it is a straight-forward task to (a) perform an energy balance and (b) calculate average cycle efficiency for any prescribed duty cycle. By changing the numerical values of parameters in the component models, it is possible to ascertain the effect of changing components on the actual physical system. Other types of analyses which rely on performance simulation are described below.

OPERATIONAL SEVERITY ASSESSMENT

It is well known that the life of both metallic and non-metallic components is influenced by stress cycling. In most fluid power components the stress levels are directly proportional to pressure levels. Traditionally cyclic endurance tests have utilized a constant amplitude-constant-frequency wave form for component tests. By using the pressure time-histories obtained by simulation, it is possible to synthesize more realistic wave forms for endurance testing. The effect

of changing component parameters on operational severity has been examined in another study [Ref. 20].

THERMAL APPRAISAL

The heat generation and dissipation characteristics of a hydraulic system depends upon the duty cycle and the operational environment. Calculation of local temperatures, at valves, actuators, etc., requires a knowledge of local pressures and flows, i.e., can only be done after performance simulation. The equations for thermal appraisal, i.e., the thermal model of the system, predict the temperature of, and the heat flow rates from, different components. The steady-state thermal model portrays the behavior of a system when the heat rejection rate equals the heat generation rate and temperatures in various parts of a circuit are constant, whereas the dynamic thermal model depicts the behavior when the hydraulic components, including the system fluid, are absorbing or releasing part of the generated/dissipated heat. Some types of equipment may be operated intermittently and others may have a variable duty cycle such that neither reaches steady state operation.

PERFORMANCE DEGRADATION ASSESSMENT

Both static and dynamic simulation can be used to examine the effect of component degradation on system performance. Changes in component characteristics due to degradation are reflected by changes in parameters in component mathematical models. Thus, increase of slip flow of a pump and decrease of cracking pressure of a relief valve can

be modeled as changes in coefficients in their respective models. By iteratively simulating a duty cycle using models of "new" and "worn out" components, the effect of degradation can be ascertained.

COMPONENT DEGRADATION ASSESSMENT

Changes in component parameters can also be used for nonintrusive diagnostics. The methodology for such analysis has been outlined by Iyengar [Ref. 21]. Test data collected by a Statistical Analog, Monitor (STAM) [Ref. 5] is compared with simulated STAM profiles obtained by incorporating "worn" out components in the mathematical model. In the case of most components it is sufficient to consider the static model. However, experimental data collected on valves indicate that significant differences in dynamic behavior are manifested by new and old components [Ref. 20].

LIFE CYCLE COST ANALYSIS

The concept of appraising a fluid power system by its life cycle cost rather than the initial capital expenditure is gaining wide acceptance. The operating costs of a system can be estimated from the performance model after the vehicle mission, over its life-span, has been described in terms of an operational duty cycle. Apart from manpower costs, the major share of operating costs will generally be for energy. Estimation of maintenance costs needs the development of component degradation models which correlate the change in performance parameters to such factors as the contaminant exposure rate, environ-

mental conditions and operational severity. Such models are almost always empirical in nature and rely on laboratory tests, usually under accelerated conditions, to establish model parameters. Such models need to be verified by field tests in order to ensure that their predictions are reasonably accurate.

CHAPTER V

COMPUTER MODELS AND COMPONENT TEST PROCEDURES

The object of writing specifications is to convey, precisely and unambiguously what is required. Test procedures have to be written in sufficient detail to obtain information, based on which it can be decided if a particular component has met the specifications. Equipment users, including the military, have a number of reasons for having a component test program. Some of these have been outlined by Hopler [Ref. 22] as follows:

1. Provision of maximum reliability - End items don't fail in total, but as a series of component breakdowns and final reliability depends on component reliability.

2. Availability for spare parts - During replacement the original parts used are often not available within a satisfactory time frame, and the initial specification is used as a guide for spare parts procurement.

3. Establishment of minimum acceptable quality - Specifications and tests provide definitions of what level of quality is needed.

4. End item tests can't define all conditions - It is impossible to have one end item test reflect all levels and types of field exposure.

5. Life test - End item tests only partially verify each component.

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6. Control during production - This provides a base line for comparison if the components appear unsatisfactory during production line inspection.

Most of the above reasons pertain to the reliability of machinery; however, component performance tests are also needed to serve as bench marks for degradation assessment. Since few hydraulic systems are built from the ground up by one manufacturer, a large user will find himself using the same component (e.g., a relief valve) on different machinery at different locations. In the interests of interchangeability, it is highly desirable that the component specification be performance-specific rather than machine-specific. That is, it is preferable to ask for a relief valve which "passes a flow of 100 litres per minute, when the pressure differential across its ports is 17.24 MPa," rather than for one which "ensures that the pressure in the X-subcircuit of the Y-machine shall not exceed 17.24 MPa at any time."

Equipment users, interested in procuring spares can develop and use computer models to assist in efficient purchasing and stocking, provided test procedures have been drafted with such data usage in mind. The remainder of this chapter discusses the drafting of component performance specifications and test procedures so that tests furnish sufficient data to develop suitable computer models. For convenience in presentation, static performance will be treated first, and then dynamic performance. Since static tests can be viewed as

but a special case of dynamic tests, almost all the comments made relative to the former apply to the latter.

STATIC TEST PROCEDURES

For some components geometrical parameters and manufacturer's data are sufficient to develop a static mathematical model and testing is required only to verify the accuracy of the information furnished by the vendor. Thus hydraulic cylinders, accumulators, hoses and tubing can often be modeled from manufacturer's technical brochures. For some components, e.g., valves, pumps and motors, the graphs and charts provided by the vendor may be adequate to develop computer models. Quite often, however, the information provided in vendor's technical brochure is insufficient. Few of them for instance, indicate the effect of changing fluid viscosity on pump and motor efficiency and leakage, or cylinder drag. Hence, tests have to be performed, by either the equipment user, the component vendor, or a third party, to ascertain these effects. The fundamental point to be kept in mind while drafting the test procedures is that the tests be repeatable and reproducible. Repeatability implies that if a prescribed test is performed on a given component at different times on the same test set-up, the results agree, within the specified tolerance limits. Repeatability is thus a measure of the accuracy with which test parameters, e.g., temperature, flow rate, etc., are controlled by the test stand as well as the accuracy of the instrumentation.

Reproducibility requires that a test performed in one laboratory yield the same test results as that in another laboratory which may not have an identical test set-up. Reproducibility has to be assured, in general, by describing the test stand performance in sufficient detail in the test procedure.

Quite often, when the suitability of a component for a specific machine is being examined, it is sufficient to ascertain the performance characteristics of the component over a limited operational range. Thus, if a pump is expected to operate only between 6.9 MPa and 10.3 MPa, on a specific machine, there may be no need to appraise its efficiency beyond this range, even though the manufacturer may have rated it at 13.8 MPa. However, if the same component is used on different machines or there is a likelihood that a hydraulic circuit using it may be modified for special purposes, it is advantageous to test it over its entire range of operation. Additionally, if all significant parameters are measured and recorded, the data will be usable to develop an empirical mathematical model of the component. It is important to emphasize that all significant parameters be measured and recorded, even though they may be constant during the test. Commonly overlooked parameters are suction pressures for pumps, downstream or tank port pressures for valves, and case drain line pressures for pumps and motors. Hydraulic fluid properties, e.g., viscosity, viscosity index, and density, should be spelled out, or an appropriate citation made to ASTM or other standards, in preference

to indicating a commercial brand name. Test circuit schematics and component mounting arrangements should be explicitly included in the test procedure. Though they should not be formatted too rigidly, the test report should be required to include circuit schematics and sketches so that the set-up can be duplicated elsewhere. Particular care should be taken in specifying where transducers for flow, pressure and temperature should be located.

DYNAMIC TEST PROCEDURES

The importance of assessing the dynamic performance of fluid power components has hitherto been restricted to a few applications. These are generally characterized by the use of a servo-valve or a variable displacement pump. In some instances, the importance of dynamic response of relief valves has been recognized as evidenced by procurement specifications which call out a "response time."

The fundamental objective in drafting test procedures of evaluating dynamic performance of components should be to acquire sufficient information to develop reasonably accurately their mathematical models. Such models will of necessity be empirical or semi-empirical since few manufacturers would divulge constructional details of their products. Since the laboratory bench test set-ups can rarely duplicate the actual system in which the test component is intended to be used, variables to be monitored should be carefully selected and appropriately monitored. Thus, it is not sufficient to specify that "a solenoid valve

be suddenly opened to impose a change in flow rate from a 10 ℓpm to 100 ℓpm through a relief valve." The word "suddenly" should be quantified, preferably by indicating how fast the flow should actually change. It must be emphasized that physical transients observed in a test set-up do not directly indicate the level of transients that may be experienced in the actual system. This is because such transients are affected by not only the test component but also other system parameters. Thus, the maximum transient pressure when a relief valve is subjected to a sudden change in flow will depend on the elasticity of the upstream lines and the volume of upstream fluid, and these may be different for the test set-up than the actual system. Consequently, a specification which says that a "valve response shall limit surge pressure to 20.7 MPa and a pressure rise rate to 170 MPa/sec." is incomplete unless the test set-up specification includes the upstream volume and elasticity.

As a working rule, all port variables of a component (two to each energy port) should be monitored during a dynamic test. All input quantities (e.g., spool movement and solenoid actuation) should also be recorded.

The accuracy of instrumentation and data recording is a crucial factor in the development of mathematical models from dynamic test data. The desired dynamic response of instrumentation and the data-sampling rate, in the case of digital recording equipment, must form

an integral part of the test procedure. In the case of most transducers and recording equipment, these specifications can be given in the frequency domain. In some cases, it may be necessary to specify the use of filters to remove noise and extraneous signals from useful data.

Some components (e.g., relief valves) are very sensitive dynamically to the components to which they may be attached. Such components are said to be closely coupled with other circuit components. In such instances, it is often advantageous to test a subsystem comprised of two or more components than to test them individually. Thus, if a relief valve fits in a cavity in a block of spool valves, it might be more convenient and desirable to test the relief valve when it is installed in the valve block than in a special housing. However, a dynamic test performed under such circumstances may not shed much light on how the relief valve would behave in another valve block.

Since most hydraulic system components exhibit nonlinear behavior and are multi-input, multi-output by nature, the selection of inputs to be used for dynamic tests is not always easy. A few general guidelines can, however, be offered:

1. Vary all inputs over the entire operating range of the component.

2. Vary inputs as quickly as the component is likely to experience in actual practice.

3. If the component is likely to operate around a steady-state value for long periods, obtain the response for small inputs around this operating point.

4. If actual inputs on the machine are available, duplicate them for bench tests.

In closure it can be said that the drafting of test procedures and specifications for hydraulic components is still very much an art, and past experience with similar systems and components is invaluable. Too often, however, the experience gained on one type of system is misapplied to others, resulting in disasterous consequences. Another ever-present danger is that of "gold-plating," (i.e., including meaningless specifications just to be "on the safe side").

Hopefully, as more and more systems are systematically analyzed prior to construction, and their predicted behavior compared with actual performance, system designers will be able to draft general test procedures for components rather than tailoring it for a narrow class of systems. Such will be the case only if the information generated by the tests is presented in the form of mathemaical models which can be used for synthesizing the mathematical model of any system of which the component may form a part.

CHAPTER VI

CONCLUSION & RECOMMENDATIONS

Fluid power system designers and machine users have until recently relied almost entirely on technical brochures and product catalogs to acquire information on components being incorporated in a system. Typically these data are given as verbal descriptions or in the form of graphs. As explained in Chapter II, it is possible to develop mathematical models from such data. Since the format for data presentation and the amount of information varies widely from manufacturer to manufacturer, it is not possible to outline a standard procedure for extracting a mathematical model from manufacturer's brochures. It is generally preferable to use test data on components and subsystems to develop reliable empirical and semi-empirical mathematical models.

As hydraulic systems become more and more sophisticated, it becomes harder to perform, by hand, pressure, flow and temperature calculations. Hence, computerized data analysis and simulation become cost-effective, especially when they can be substituted for extensive testing. Even though the fluid power analyst need not be a computer programmer, it is advantageous for one to be aware of computer software suitable for simulation, and the type of results that can be expected. Chapter III has presented a comprehensive survey of computer programs for static and dynamic simulation of fluid power systems. The general purpose simulation languages require some knowledge of electrical analogies and

systems theory on the part of the user, while the special software either requires generally unavailable design details or is applicable only to a class of systems. As the fluid power industry expands usage of computer simulation it is expected that more effort will be devoted to the development of software applicable to most, if not all, fluid power systems.

Component performance models can be used to predict the hydraulic variables, i.e., pressure and flow in a given circuit, as the machine is operated in a prescribed fashion. Additionally, they can be used to perform an energy balance, estimate average efficiency, and calculate temperatures in various points of a circuit. By allowing the parameters in the performance model of a component to vary, it is possible to examine the effect of component degradation on system performance. The methodology for such appraisal has been sketched in Chapter IV.

The system designer, in contra-distinction to the component designer, usually does not have access to the internal constructional details of components. Hence, he needs to develop component models from test data. Unless test procedures are clearly and unambiguously described and all relevant test parameters included in the report, it is difficult to use test results for developing component models. Chapter V presented guidelines for drafting test procedures to ensure that test data is usable for model development and verification.

In order to improve the reliability of mobile hydraulic equipment in a cost effective manner, the following recommendations are offered.

1. All test procedures should be drafted so that tests are component-specific rather than machine specific. This will not only facilitate interchangeability, but also permit the equipment user to modify existing machines for special missions.

2. Test procedures should require that all significant physical quantities, pertaining to a component, should be measured and recorded, even though they may not vary. Examples of such variables are fluid density, ambient temperature, and in some hydraulic systems, tank port pressure.

3. In the course of a test physical variables, e.g., flow rate, pressure, temperature should be varied over the potential range of operation of the component rather then merely what would be experienced in a specific application. Thus, pump efficiency tests should be performed over a wide range of temperatures than at just one temperature, as is currently the case with most vendors. Only by collecting test data in the above manner can empirical models be developed without resorting to extrapolation of test data.

4. Establish which parameters in a mathematical model are changed by degradation and develop mathematical relationships to describe the changes.

5. Examine the effect of component degradation on system performance by simulation and correlation with historical data.

Large manufacturers of mobile hydraulic equipment have full-time systems analysts and supporting staff engaged in computer modelling and simulation. Some have developed their own library of computer programs tailored to their product lines. Component manufacturers, who are generally smaller than end item companies, have made less use of computerized analysis. Until large-scale users of components insist on test data being acquired and presented in a standard format, the present "method" of presenting component characteristics as a haphazard collection of graphs and tables will prevail. Simulation and computer analysis has been found to be a cost effective tool in the electronics, aerospace and structures fields and there is no reason to doubt that similar results can be achieved in the fluid power industry.

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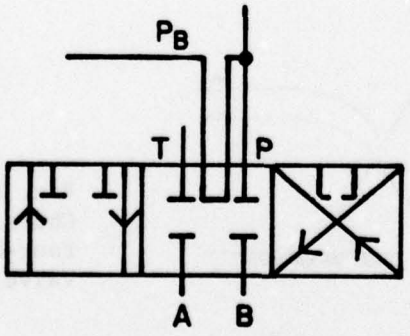
APPENDIX

LIBRARY OF MODEL FORMS

In the following pages, we present semi-empirical model forms for a number of commonly used hydraulic components. These forms are such that by conducting appropriate bench tests the parameters for a specific design can be identified using standard numerical techniques. A rudimentary knowledge of the constructional details of a component are used to evolve the model form. However, knowledge of internal constructional details is neither used nor desirable if the models are to be kept simple. Individaul system designers may refine these models by exploiting their knowledge of internal constructional details or behavior observed in similar designs. The forms presented here are not sacrosanct and may be replaced by other forms if the latter are expected to result in better fit to data. Since most components are multi-input and nonlinear, the implicit form of representation has been chosen.

COMPONENT: Open-Center Four-Way Valve

SYMBOL



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NOMENCLATURE: (See Fig. A-1)

P_s	Supply Pressure
Q_s	Supply Flow
P_A	Port 'A' Pressure
Q_A	Port 'A' Flow
P_B	Port 'B' Pressure
Q_B	Port 'B' Flow
P_T	Tank Port Pressure
Q_T	Tank Port Flow
x	Spool Displacement
a_1	Effective Metering Orifice Area, P to A (See Fig. A-1)
a_2	Effective Metering Orifice Area, P to B
a_3	Effective Metering Orifice Area, A to T
a_4	Effective Metering Orifice Area, B to T
a_5	Effective Metering Orifice Area, P to T
k	Orifice Flow Coefficient
Q_5	Bypass Flow

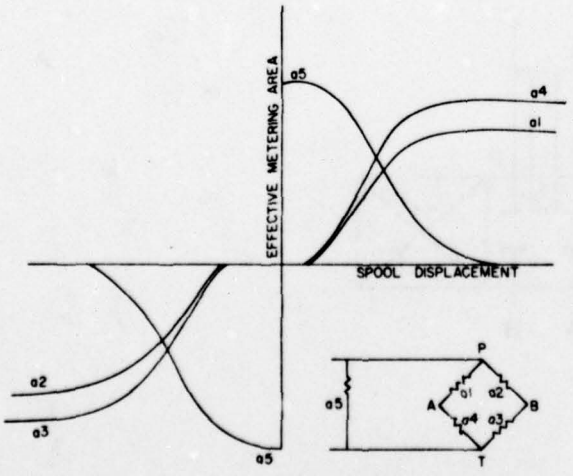


Fig. A-1. Metering Characteristics of a Four-Way Open-Center Valve.

STATIC MODEL:

For $x > 0$

$$0 = -Q_S + Q_A + Q_5 \tag{1}$$

$$0 = -Q_A + ka_1(x)\ (P_S - P_A)^{1/2} \tag{2}$$

$$0 = -Q_5 + ka_5(x)\ (P_S - P_T)^{1/2} \tag{3}$$

$$0 = -Q_B + ka_4(x)\ (P_B - P_T)^{1/2} \tag{4}$$

$$0 = -Q_T + Q_B + Q_5 \tag{5}$$

For $x < 0$

$$0 = -Q_S + Q_B + Q_5 \tag{6}$$

$$0 = -Q_B - ka_2(x)\ (P_S - P_B)^{1/2} \tag{7}$$

$$0 = -Q_5 - ka_5(x)\ (P_S - P_T)^{1/2} \tag{8}$$

$$0 = -Q_A - ka_3(x)\ (P_A - P_T)^{1/2} \tag{9}$$

$$0 = -Q_T + Q_A + Q_5 \tag{10}$$

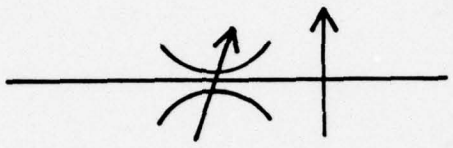
Equations (1) through (5) would be used for positive spool displacements and (6) through (10) in case of negative spool displacements. Note that, at any time, only two of the four orifice forming the Wheatstone bridge are open, in addition to the bypass orifice, a_5 (Fig. A-1). The metering characteristics shown in Fig. 1 may be obtained from component drawings and specifications, or from test data on an actual valve. For simplicity, it may be sometimes preferable to replace the curves shown in Fig. A-1 by straight line segments and develop analytical expressions for the metering orifices in terms of the spool displacement. Given a spool position, the metering areas are always known and, consequently, there are nine variables interrelated

in five equations—hence, four variables have to be furnished as inputs to the model.

DYNAMIC MODEL: Same as the static model.

COMPONENT: Pressure-Compensated Flow-Control Valve

SYMBOL:



NOMENCLATURE:

P_S	Supply Pressure
P_T	Tank Port Pressure
P_d	Load Pressure
Q_S	Supply Flow
Q_d	Load Flow
Q_T	Tank Port Flow
A_1, A_2, k_o, k_1, b, x_o	Empirical Coefficients

STATIC MODEL:

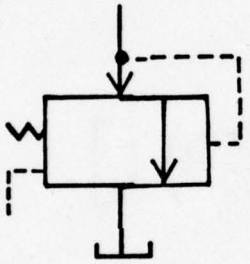
$$0 = -Q_S + Q_d + \frac{k_o}{b}\left[P_S(A_2 - A_1) + \left(\frac{Q_d}{k_1}\right)^2 A_1 - bx_o\right](P_S - P_T)^{1/2} \qquad (1)$$

$$0 = -P_d + P_S - \left(\frac{Q_d}{k_1}\right)^{1/2} \qquad (2)$$

Equation (1) relates the pressure and flow at the supply port and the bypass port pressure, while Eq. (2) is an expression for the load pressure in terms of the load flow.

COMPONENT: Pressure-Relief Valve

SYMBOL:



NOMENCLATURE:

P	Pressure Differential Across Valve
P_{up}	Upstream Pressure
P_T	Tank Port Pressure
Q	Flow Rate Through Valve
ρ	Fluid Density
A_o	Effective Metering Area
k_1, k_2, k_3	Coefficients in Semi-Empirical Models

STATIC MODEL:

$$0 = -Q + \frac{k_1 (P_{up} - P_T) - k_3}{\sqrt{\rho/2} - k_2 (P_{up} - P_T)} \quad \sqrt{(P_{up} - P_T)} \tag{3}$$

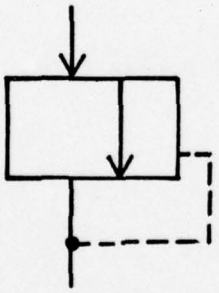
Equation (3) furnishes an expression for the flow through the valve, which is modeled as a variable area orifice, whose metering area is controlled by a moving element kept in balance by the pressure differential and spring forces. The equation is, of course, valid only for positive flow rates and positive pressure differentials.

DYNAMIC MODEL:

$$0 = -\dot{P} + \frac{2\sqrt{P}}{Q}\left[\dot{Q}\sqrt{P} + \left(\frac{Q}{\sqrt{P}} - \frac{Pk_1 + k_3}{\sqrt{\rho/2}}\right)\frac{\dot{P}}{k_3}\right] \tag{4}$$

COMPONENT: Pressure Reducing Valve

SYMBOL:



NOMENCLATURE:

P_{up}	Upstream Pressure
P_d	Downstream Pressure
ρ	Fluid Density
Q	Flow Rate Through Valve

A_o Effective Metering Area

k_1, k_2 Coefficients in Semi-Empirical Model

STATIC MODEL:

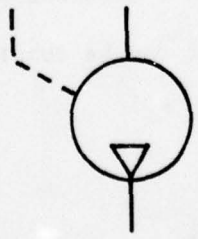
$$0 = -Q + A_o \sqrt{(P_{up} - P_d)\ (2/\rho)} \tag{5}$$

$$0 = -A_o + k_3 - k_1 P_d \tag{6}$$

Equation (5) furnishes an expression for the flow through the valve, which is modeled as a variable area orifice. Equation (6) describes the force balance on the moving element controlling the variable area orifice. This equation is, of course, valid only for positive values of A_o. Generally P_{up} is kept constant by an upstream relief valve.

COMPONENT: Fixed-Displacement Pump

SYMBOL:



NOMENCLATURE:

P_S	Pump Outlet Pressure
Q_S	Pump Outlet Flow
D_p	Pump Displacement/Revolution
N_p	Pump Rotational Speed
T_p	Torque Input to Pump
P_T	Pump Inlet Pressure
T_{cp}	Constant Torque Loss
R_p	Torque Loss Coefficient
k_p	Leakage Coefficient

STATIC MODEL:

$$T_p = D_p \frac{(P_S - P_T)}{2\pi} - R_p N_p - T_{cp} \tag{7}$$

$$Q_S = D_p N_p - k_p P_S \tag{8}$$

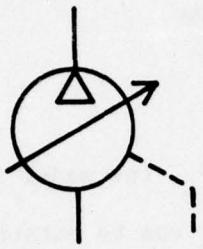
Equation (7) furnishes an expression for the input torque to the pump, while Eq. (8) yields a value for the outlet flow in terms of the outlet pressure. If one is not interested in the torque input to the pump, Eq. (8) by itself can be considered to be the model.

DYNAMIC MODEL:

Same as Static Model.

COMPONENT: Pressure-Compensated Variable-Displacement Pump

SYMBOL:



NOMENCLATURE:

P_S	Pump Outlet Pressure
Q_S	Pump Outlet Flow
D_p	Maximum Displacement
N_p	Pump Rotational Speed
T_p	Torque Input to Pump
T_{cp}	Constant Torque Loss
R_p	Torque Loss Coefficient
P_c	Compensator Set Pressure
D	Pump Displacement (Pressure Dependent)
P_T	Pump Inlet Pressure
k_c	Compensator Coefficient
k_p	Leakage Coefficient

STATIC MODEL:

$$D = D_p \quad \text{if } P_S < P_c \tag{9}$$

$$D = D_p - (P_S - P_C)k_c \quad \text{if } P_S > P_c \tag{10}$$

$$T_p = D_p \frac{(P_S - P_T)}{2\pi} - R_p N_p - T_{cp} \tag{11}$$

$$Q_S = D_p N_p - k_p P_S \tag{12}$$

Note that if P_S is less than P_C, the model is exactly the same for the fixed-displacement pump, D_p can be either positive or zero.

DYNAMIC MODEL

The dynamic model for a pressure-compensated variable-displacement pump has to generally take into account the following items:

(i) the inertia of the yoke (sometimes called the hanger) and stroking mechanism

(ii) flow gain of the compensator valve

(iii) fluid compressibility and inertia effects

(iv) damping due to viscous friction, nonlinear orifices and sometimes flow force effects

Since the stroking mechanism, feedback paths and compensator valve flow paths can differ from design to design, it is not possible to develop a single set of equations which would apply across the board to all designs.

A typical model would consist of equations for describing:

(a) the force balance on the stroking actuator

(b) the flow balance for the compensator valve as well as the pump

Both types of equations will be differential equations, since the inertia of the stroking mechanism and the compressibility of the fluid are involved. A well-documented model is contained in Ref. [1.1].

SECTION VI

NONDESTRUCTIVE CONTAMINANT SENSITIVITY

PROJECT PERSONNEL

Dr. E. C. Fitch, Program Director

Dr. R. K. Tessmann, Program Manager

Mr. J. M. Howsden, Project Engineer

Mrs. Janice Dobson, Ferrography Analyst

PREFACE

The section presents a detailed account of the project activities in the area of nondestructive contaminant sensitivity testing. A rigorous analytical wear model for a hydraulic pump is developed. This model is verified through the use of the results of over 24 pump tests. The appendixes of this section contain all of the data generated in the activity reported here.

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CHAPTER I

INTRODUCTION

The evolution of high pressure, high performance hydraulic systems, accompanied by the inflationary economic spiral, has focused considerable attention upon the life and reliability of such systems. The initial cost of modern hydraulic components is high and this cost added to that of maintenance, downtime, and repair has created a great demand for hydraulic systems which can survive the rigors of field operation for long periods of time. The field service life of hydraulic components is assessed through their performance. That is, a hydraulic pump, for example, is considered "worn out" when it no longer provides sufficient flow at the required operating pressures. Hence, to be useful, any study which is directed toward assessing the potential life of a hydraulic component must be based upon the performance of the components.

Wear is one of the most costly phenomenon which occurs during the use of devices in the world today. The U.S. Navy has reported [1] that the cost of wear control in aircraft and surface ships is approximately two-thirds the cost of the fuel. In many wear processes, especially those occurring in oil wetted components, abrasive particles serve to accelerate the deliterious attack upon critical internal surfaces. In some cases, engineering have learned to harness this phenomenon for the benefit of industry. For example, various manufacturing processes (such as grinding, lapping, sanding, etc.) make good use of abrasive particles in controlling the rate of material removal and the texture of the finished

surface. However, when such processes are undesirable, as is the case in hydraulic components, abrasive particles become contaminants and their effect has fostered an area of engineering science appropriately named contaminant wear.

Three basic modes of contaminant related failures have been defined for hydraulic system components [2] [3]— transient, catastrophic, and performance degradation. Transient failures are characterized by a temporary malfunction, such as an excessive pressure overshoot or a momentary hesitation in response. Such unsatisfactory performance is caused by the presence of contaminant particles lodged in critical clearance spaces of a component causing increased drag forces or a temporary change in the relative configuration between mating parts. Catastrophic failures, on the other hand, occur suddenly like the transient failures but are of a more permanent nature. This type of failure may be typified by a complete locking of moving parts (such as directional valve spools or relief valve poppets)— a condition normally referred to as contaminant lock [4]. In many components, contaminant particles can plug small orifices which control critical hydrostatic balances. When this occurs, the balance is lost, and an almost immediate catastrophic failure will follow.

While both transient and catastrophic failures certainly reduce the life and reliability of hydraulic systems, it is almost impossible to treat them as wear processes. Performance degradation is the only basic contaminant related failure mode which predominately occurs as the result of a wear process. Such a failure results when the performance

of a hydraulic component is impaired to such an extent that it must either by repaired or replaced. The characteristic feature of this failure mode is the gradual but persistent deterioration of critical surfaces within the component. In the case of hydraulic pumps, a prolonged attack by particulate contaminants entrained in the system fluid will result in an enlargement of the leakage paths inherent to the pump. The larger paths will accommodate an increased leakage flow, which will be reflected in a lower output flow from the pump. In addition, the wearing away of these critical surfaces will produce large quantities of metallic debris which adds to the particulate matter already entrained in the system fluid.

The contaminant protection level required by a system to minimize contaminant wear will be dictated by the component which exhibits the greatest sensitivity to entrained particles. The three most critical components in a hydraulic system (from a performance standpoint) are the pumps, valves, and cylinders. Normally, valves and cylinders are characterized by low relative velocities between sealing surfaces and/or elastomeric seals. However, the sealing of high and low pressure areas of most hydraulic pumps is typified by extremely close tolerances and high relative velocities. Therefore, the hydraulic pump is the most likely candidate for the greatest component contaminant sensitivity. Field reports, other studies, and experience bear out this conclusion. In addition, the widespread use of the gear type hydraulic pump in the fluid power industry makes it an obvious choice for contaminant wear studies which will benefit hydraulic system users.

The configuration of the leakage paths in a hydraulic gear pump is complex and the contaminant wear process is extremely difficult to simulate without actually operating the pump in a controlled contaminant environment under realistic conditions. This means that the evaluation of the wear process must be made while the pump is operating in a circulating hydraulic system. Basically, wear analysis methods which could be used in such a situation can be classified as either intrusive or non-intrusive. Intrusive techniques rely upon metrological evaluations (direct measurements of surface changes), while non-intrusive approaches include performance degradation measurements and/or wear debris evaluation. While intrusive methods may be of some value to the pump designer in usually observing and quantifying the damage done, they have not been widely employed mainly due to the problems of disassembling the pump, making precise microscopic measurements, and attempting to relate these with the real world of performance. The non-intrusive method employing direct performance monitoring has gained considerable popularity for evaluating contaminant wear in hydraulic pumps. While this approach overcomes the problems associated with the intrusive method, it has distinct disadvantages of relying upon gross component destruction to produce sufficient surface deterioration for accurate detection and appraisal.

Wear debris analysis is a non-intrusive method which has been applied much more to lubrication systems than to hydraulic systems in the past. Spectrometric analysis of the fluid from an oil wetted component produces information relative to the elemental composition of the generated debris. However, the equipment needed to conduct spectrometric analysis is extremely sophisticated and expensive. In addition, some recent studies [5] indicate that such

analysis may not produce all of the essential information relative to wear processes. A relatively recent technique involves the isolation and evaluation of wear debris entrained in the fluid of a lubrication or hydraulic system based upon a technology called Ferrography [6]. Basically, the Ferrographic Oil Analysis System employs a flowing stream in which entrained particles are subjected to a combination of magnetic, gravitational and fluid drag forces. By design, the magnetic field is the most dominate and causes the separation of magnetically active particles from the fluid and environmental contaminants [7]. Once isolated, the amount of debris is measured to determine the severity of the wear process, while the size, shape, structural appearance, etc. of the debris particles provide valuable information relative to the most influencial wear modes.

The capability of the Ferrographic technique to separate magnetically active wear particles from other particulate contaminants which are present in the oil of a hydraulic system make this technique uniquely suited to contaminant wear studies. In such investigations, wear is induced by means of a non-magnetic test contaminant. Without a technique to separate the generated debris (which is, in general, magnetically active) from the introduced test contaminant, debris analysis of contaminant induced wear would be virtually impossible. Previous investigations [8] [9] using the Ferrographic Oil Analysis System procured by the Fluid Power Research Center indicate that the method is potentially capable of measuring the debris generated during a contaminant sensitivity test on a hydraulic gear pump.

The objectives of this study is to show that Ferrographic Analysis is capable of not only measuring the debris generated in a standard pump

contaminant sensitivy [10] test, but is also capable of producing useful results at contamination level much lower than that specified in the standard test. A further objective of this work is to demonstrate the correlation between Ferrographic data and flow degradation in pump when exposed to a controlled contaminant environment. This will be accomplished by developing a simplified, lumped-parameter model for the pump leakage paths and showing that changes in these path dimensions will produce both debris and flow degradation. Thus, the use of such a pump model provides a characteristic clearance as an intermediate parameter which is employed in formulating the desired relationship.

The ultimate benefits which can be derived as a result of this project are as follows:

1. More contaminant wear information can be obtained from a single pump because the increased sensitivity of the Ferrographic analysis method will produce wear data with little or no measurable destruction of the pump.
2. Establishment of a pump contaminant sensitivity procedure which would be conducted at a contaminant level closer to that of a actual fluid power system. The magnitude of the contamination level is the only acceptable criticism of the standard pump contaminant sensitivity test.
3. Provide more information for pump and filter selection. Since the pump is damaged very little during the test, numerous tests can be run on the same pump, thus provideing a broad spectrum of data.

4. Provide background needed to advance a powerful wear assessment technique which can be utilized in the contaminant sensitivity appraisal of other critical hydraulic components. Due to the great diversification in application of some components, such as valves, it is extremely difficult to select a performance parameter for use in contaminant wear assessment which is useful in all applications. However, since all performance degradation caused by a contaminant attack in any component can be related to debris generation, the Ferrographic method of wear debris analysis can be used successfully on a wide variety of hydraulic components.

In order to accomplish the objectives of this project, several pump tests will be conducted. The pumps are from the same lot of the same manufacturer and are gear-type design. The test work will vary contamination level, particle size range, pressure, speed, and test sequence. The next chapter discusses the concepts associated with contaminant wear and outlines the results of previous investigation. Chapter III concentrates on a discussion of the Ferrographic Analysis technique and its capabilities and limitations. The relationship required to transform Ferrographic data to flow degradation are developed in Chapter IV, while Chapter V is concerned with the experimental results from the pump tests. Chapter VI presents a summary of the project along with specific conclusions resulting from the work.

CHAPTER II

PREVIOUS INVESTIGATIONS

The interrelated processes of friction and wear, along with the influences of lubrication, have been recognized since ancient times [11]. The use of lubricants by the Egyptians in the transport of large building blocks probably reduced the coefficient of friction and, hence, the labor force by about fifty percent. The early attempts to develop rolling-element bearings in Greek and Roman times is indicative of a desire to replace sliding by rolling contact and thus reduce friction and wear in machinery. Many of the changes in bearing materials recorded were beneficial in decreasing friction, although the main incentive responsible for the change was probably a desire to reduce the excessive wear of vital machine parts. In some instances, as in the case of the early wheeled vehicles and the potter's wheel, leather seals were employed. This indicates that these early engineers recognized the deliterious effects of contaminant wear due to free particles entrained in the bearing lubricants.

In general, scientists have divided the phenomenon of wear into several separate types [12] [13]. These wear mechanisms were given by Rabinowicz [13] as follows:

1. Adhesive or Galling Wear
2. Abrasive or Cutting Wear
3. Surface Fatigue
4. Erosive Wear
5. Corrosive Wear
6. Cavitation Wear

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It is common practice for laymen in the area of contamination control to equate abrasive wear and contaminant wear. This does not, however, provide a clear picture of the phenomenon called contaminant wear. Abrasive wear is normally divided into processes termed two-body and three-body [14]. Two-body abrasion refers to the case where there are only two surfaces involved and micro-cutting takes place due to surface asperities. Three-body abrasion is meant to imply the presence of a third member in the process (namely, a defiling particle). Obviously, contaminant wear does not emcompass that process called two-body abrasion. Actually, contaminant wear is the result of two types of wear acting simultaneously— three-body abrasion and erosion, where erosive wear is characterized by the collision of free moving contaminant particles with the critical component surfaces.

The most comprehensive studies on three-body abrasion has been reported by Rabinowicz et al [15]. The work by Rabinowicz was prompted by earlier tests conducted by Toporov [16]. Rabinowicz utilized a specially designed abrasive wear machine to produce results relative to three-body abrasion using various exposure times, material hardness, and abrasive grain size. From this study, it was concluded that the prevailing phenomenon was similar to those of two-body abrasion using metal specimens sliding against abrasive-covered paper. In comparing his results to those obtained in two-body abrasion tests it was found that abrasive wear rates during three-body abrasion are about ten times less during two-body abrasion.

The results of two-body abrasion using abrasive covered paper were reported by Nathan and Jones [17]. The paper presented by Nathan and Jones contended that there were many inconsistencies in the abrasive wear of metals

as determined by different authors which necessitated a detailed investigation to determine the relationships existing between the abrasive wear of metals and the experimental parameters which influence abrasive wear. The conditions studies by Nathan and Jones included:

1. Applied load
2. Length of abrasive path
3. Mean diameter of abrasive particles
4. Velocity of the abrasive surfaces
5. Different types of abrasive particles
6. Temperature of the environment
7. Absolute humidity of the atmosphere

Of particular interest here is the results obtained by Rabinowicz using various particle sizes (abrasive grain sizes) and the results reported by Nathan and Jones with various applied loads, diameter of abrasive particles and velocity of the abrasive surfaces. Both of these investigative teams concluded that abrasive wear rates increased linearly up to a particle size of 70-80 micrometres. Above that size Rabinowicz contended that wear rates were independent of abrasive grain size while Nathan and Jones showed a greatly reduced dependency. In considering the conclusion that abrasive wear is linear with particle size below 80 micrometres, it must be noted that both of the referenced investigators used only two particle size values below that limit and evaluation of the curve fits presented indicate a rather poor correlation to this linear representation. In fact Nathan and Jones stated that the abrasive wear obtained for abrasive sizes of 35 and 45 micro-metres were less than the expected values for the linear relationship.

The experimental data acquired by Nathan and Jones using different applied loads indicate that abrasive wear increases with increasing load. In addition, the results obtained when the relative velocity of the wear surfaces was varied showed essentially no effect due to the speed parameter. Thus, based upon the data presented by both Rabinowicz and Nathan and Jones, the curves shown in Figure 1 should generally describe the results of a contaminant wear test on a hydraulic pump. In this figure, pressure 1 is greater than pressure 2 and it has been assumed that applied load as presented by Nathan and Jones would equate to pressure in a hydraulic pump. Also, since the rotational speed of the pump can be equated to the relative velocity of the wearing surfaces, there should be no influence of speed in contaminant wear of a pump.

The erosive wear phenomenon, which is believed to be an integral part of the contaminant wear process, has been investigated by numerous researchers [18] [19] [20] [21] [22] [23]. In general, it has been concluded that the volume of material removed from a surface due to erosion is a function of the mass of the impinging particles, the particle velocity, the threshold of deformation for the wearing surface and the amount of energy needed to remove one unit volume of material. This relationship can be written as follows:

$$V = \frac{1}{2} \frac{M[v - k]}{E}$$

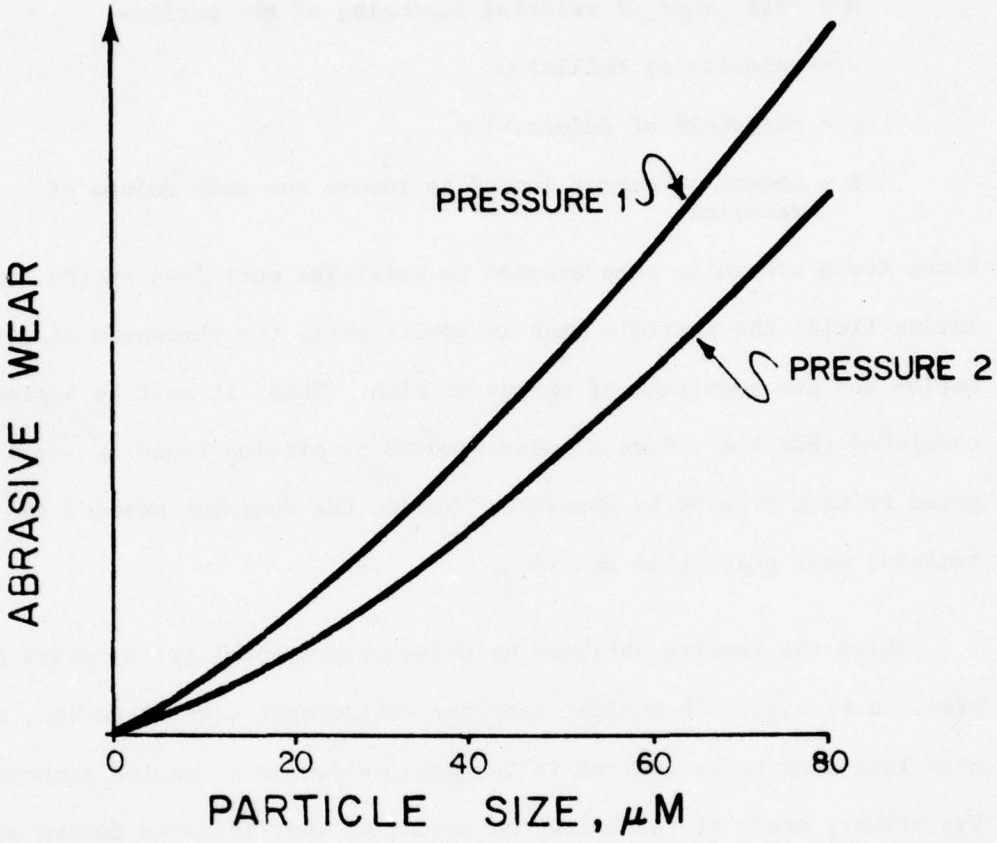


Figure 1. Expected Results for Contaminant Wear Tests on Hydraulic Pumps (Adapted from References 15 and 17).

where: V = wear volume

M = total mass of material impinging of the surface

v = velocity at collision

k = threshold of deformation

E = amount of energy needed to remove one unit volume of material

Since for a hydraulic pump exposed to particles entrained in the circulating fluid, the particle mass is small, while the threshold of deformation and the magnitude of energy is high. Thus, it must be logically concluded that the volume of wear removed by erosion would be small compared to that removed by abrasion. Hence, the dominant mode of the contaminant wear process is abrasion.

While the results obtianed by these experimental tribologists have provided considerable insight into the contaminant wear situation, their work left much to be desired in the application to a complex component. The primary means of measuring the amount of wear incurred during any of the tests was by weighing or dimensional changes of the wearing surfaces. This is expedient and straight-forward when small test specimens are used. However, this method of measuring wear for a complete component such as a hydraulic pump is totally inadequate. To assess wear in a hydraulic pump through such intrusive means, it would be necessary to disassemble the pump after any contaminant exposure, identify the microscopic changes which have occurred and measure them. Furthermore, if the decision was made to continue testing with the same pump, it would have to be reassembled with no guarantee that the process would not have been irreversibly altered.

Another dilemma which faced hydraulic system engineers in applying the work of these fundamental wear researchers lies in the transformation of the results of basic wear tests into performance of hydraulic components. For example, the main performance parameter for a hydraulic pump is output flow. If there was a transformation which converted the abrasive wear, as shown on the vertical axis of Figure 1, into change or degradation in flow rate, the problem would not exist. However, such a transformation does not currently exist and the results reported in terms of volume of material removed from test fixtures can be used only for guidance in design. It should be noted that most of the efforts reported in the area of two-body and three-body abrasion were conducted to gain knowledge concerning grinding, lapping, etc. processes used in manufacturing. In this regard the work is invaluable.

In order to evaluate contaminant wear in a hydraulic component, it was necessary to develop a technique which provides a wear parameter that could not only be measured non-intrusively, but also one which could be related to some critical performance variable. Research work which introduced and fostered this philosophy was reported by Wolf [23] at Oklahoma State University. Wolf correctly recognized that the effect of particulate contamination was to enlarge the leakage paths of a pump. This enlargement would be accompanied by increased leakage rates which would be reflected by decreased output flow from the pump. The output flow of the pump could indeed by measured non-intrusively and was, in fact, the primary performance variable of the component. To avoid confusion with wear tests designed to evaluate wear in terms of dimensional changes or weight loss, Wolf used the

contaminant tolerance and contaminant wear to imply performance degradation as a result of contaminant action.

In the work conducted by Wolf, test pumps were subjected to various types, concentrations, and particle size increments of contaminants. The pumps were operated continuously at constant values of speed, temperature and pressure, while flow degradation was measured for each contaminant exposure. The objective of this activity was to determine the sensitivity or tolerance of a given pump to various contamination levels in order to provide a filter selection basis. In 1967, Hollinger [24] reported on the results of tests similar to those conducted by Wolf. In both efforts, it was concluded that leakage flow measurement or output flow degradation was a reasonable technique for evaluating contaminant wear and that AC Fine Test Dust classified into various particle size increments produced the most change in the leakage of a hydraulic pump.

Later, in 1970, research work by McBurnett [25] proposed a contaminant sensitivity test similar to that advanced by Wolf, except that it now incorporated a multi-passing of the injected contaminant instead of a single pass. Also, instead of using de-ironized AC Fine Test Dust (ACFTD) classified into five micrometre increments (0-5, 5-10, 10-15, etc.), McBurnett's efforts utilized particle size ranges (0-5, 0-10, 0-20, 0-30, 0-40 micrometres). These size exposures were repeated three times unless the output flow of the pump degradated by more than 30% from its initial value.

The first attempt at advancing a mathematical expression for the contaminant sensitivity of a pump was reported in 1971 by Fitch and Maroney [26]. In 1972, Bensch and Fitch [27] further modified the pump contaminant test

by incrementally subjecting the test pump to a single set of increasing particle size ranges involving a circuit shown schematically in Figure 2. The particle size ranges (0-5, 0-10, 0-20, 0-30, 0-40, 0-50, 0-60, 0-70, and 0-80 micrometres) were obtained from ACFTD and injected at a gravimetric level of 300 milligrams per litre. After each contaminant exposure, the output flow was precisely measured and normalized by the rated flow of the pump to obtain the flow degradation ratio. Figure 3 illustrates the performance degradation signatures for three typical pumps where Q_f/Q_r is the flow degradation ratio as introduced by Bensch and Fitch [27]. Through industrial guidance Fitch [28] then found that curves of flow degradation ratio versus particle size could be linearized by using log-normal model. Figure 4 shows the same typical pumps as used in Figure 3 plotted as a log-normal model. The concepts and relationships presented by Bensch and Fitch [27] and Fitch [28] were used by Fitch and Tessmann [29] to evaluate pumps in general which represents the latest advances in the contaminant sensitivity of hydraulic pumps.

The primary disadvantage of the contaminant sensitivity work which has been accomplished to date is that an extremely high contamination concentration (300 mg/l) must be used for each particle size range in order to obtain sufficient flow degradation over the spectrum of sizes. Such a concentration is higher than the majority of hydraulic pumps would be expected to encounter and results in the complete destruction of most pumps at only one operating condition (speed, pressure, etc.). In addition, the results of mechanism tests as presented by Rabinowicz et al. [15] and Nathan and Jones [17] are difficult, if not impossible, to use in the assessment

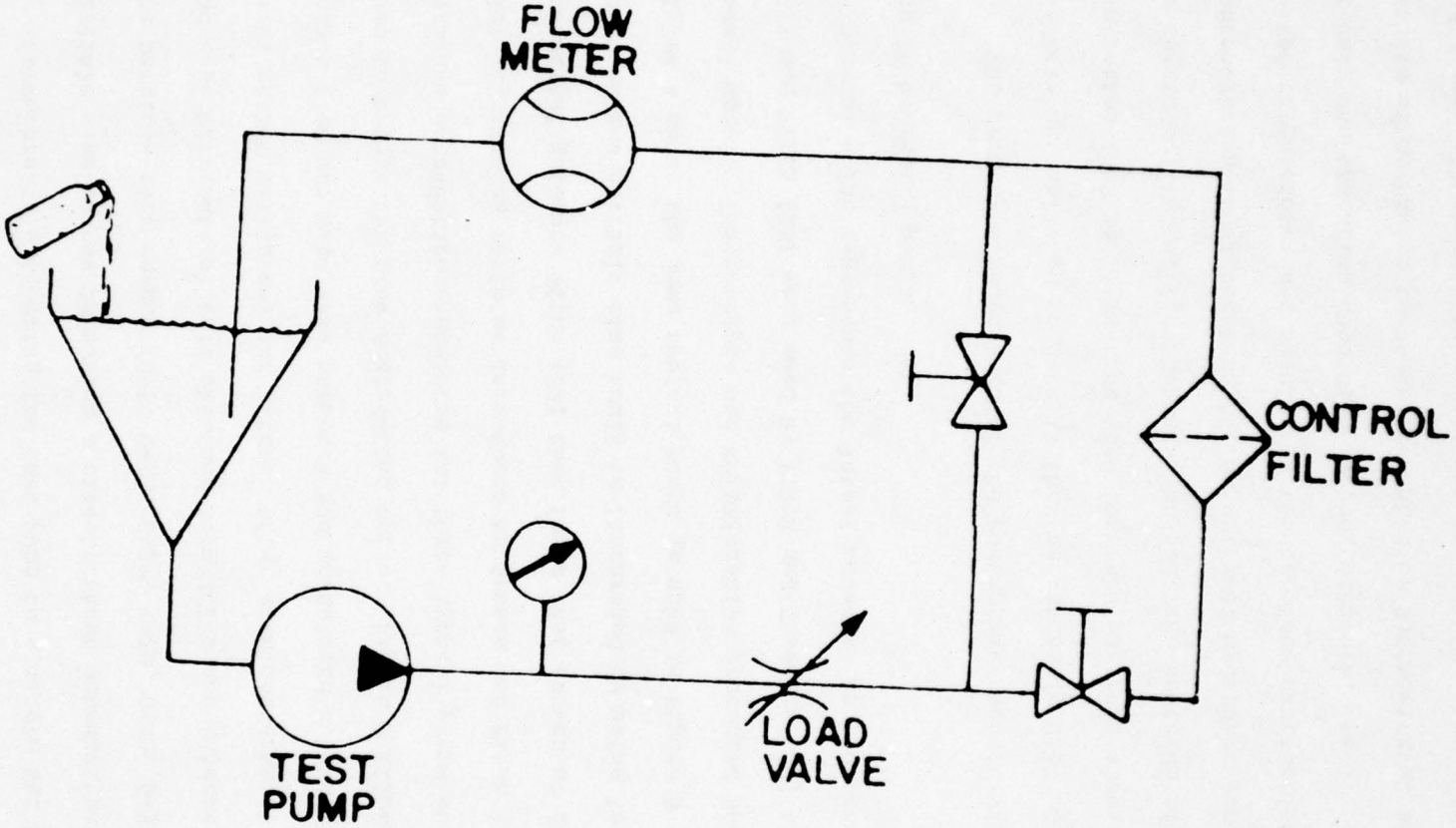


Figure 2. Schematic of Typical Contaminant Sensitivity Test Circuit

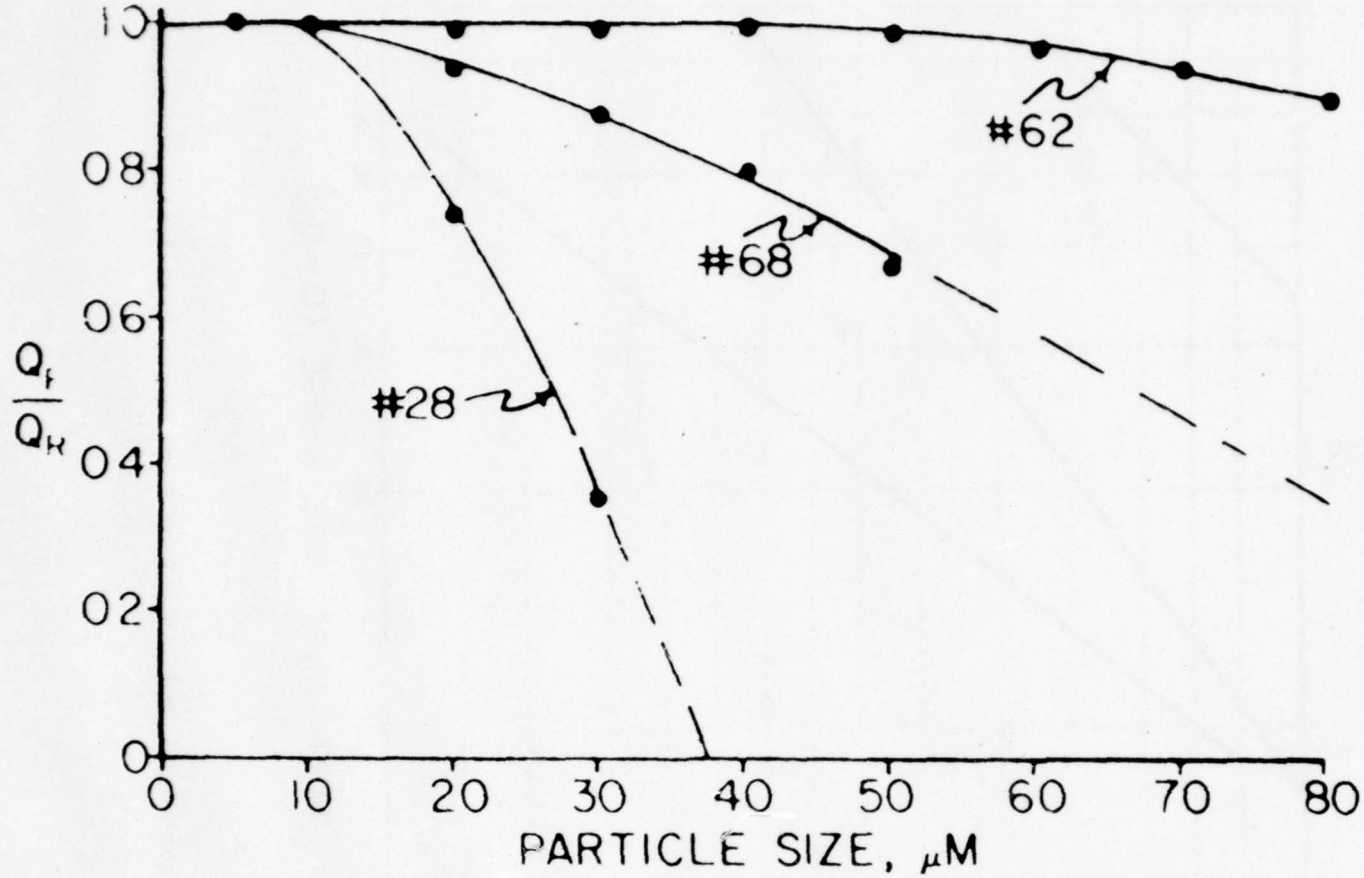


Figure 3. Flow Degradation Signatures for Typical Pumps

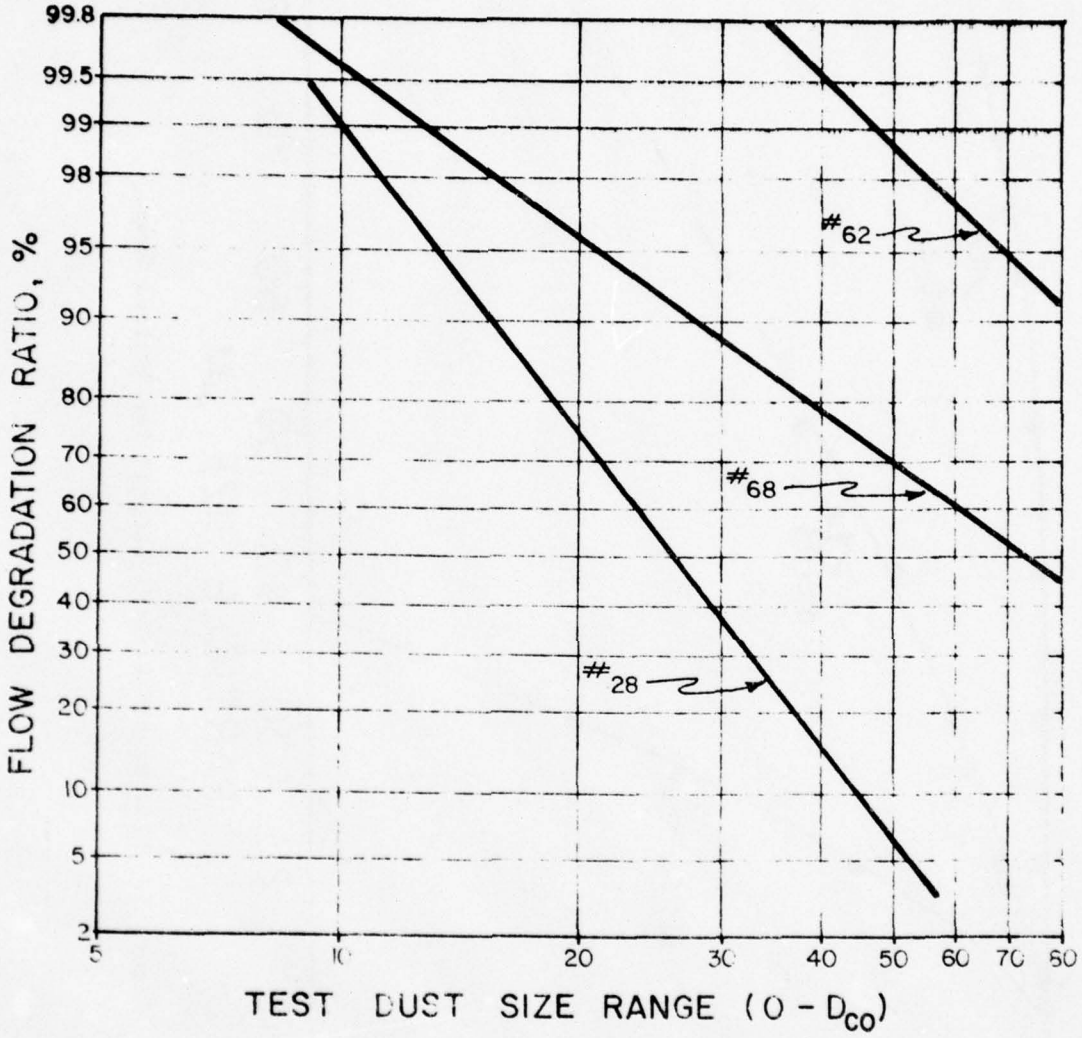


Figure 4. Log-Normal Model for Pump Performance Degradation Signatures

of performance degradation. On the other hand, direct performance degradation measurements provide little or no information relative to the nature of the contaminant attack. The answer to this dilemma is to develop a more responsive wear measurement technique which is non-intrusive and can also be related to both surface destruction and performance degradation.

The most likely candidate technique for contaminant wear analysis is associated with the evaluation of the products of the process—namely the wear debris. There are several wear debris analysis techniques which have been applied in various wear evaluation methods. The most widely used apparatus currently is the spectrometer. This device provides data on the elemental constituents of a fluid sample. While there is no doubt that fluid sampling and analysis methods are non-intrusive, there is sufficient evidence [5] which indicates that the spectrometer will not adequately assess a contaminant wear process. In fact, E. L. Falendysz, Director of Hydraulic Laboratories, J. I. Case [30] vehemently stated, "Spectrometric analysis alone is of little value in wear evaluations of hydraulic systems." In addition, Mr. B. L. Poppert, Naval Air Systems Command, AIR-340E, Washington, D. C. at a tribology workshop [31] [32] reported that his sponsored research has revealed critical inadequacies in the spectrometric methods.

Two other wear debris analysis methods which merit consideration for contaminant wear evaluation are the radioactive tracer and the neutron activation techniques. Both methods rely upon nuclear activation. In the radioactive tracer method the wearing parts are nuclearly activated prior to assembly. Then the degree of radioactivity in the circulating fluid is a measure of the amount of material which has been removed from the critical

parts. While this technique has been used on combustion engines, no record has been found of application to hydraulic components. Undoubtedly, the instrument investment compared to the component costs makes the radioactive tracer technique extremely impractical.

The neutron activation method of wear debris analysis was perfected and has been successfully applied by the Tribology Laboratory at the Research Institute for Automotive Industry (RIAI) in Hungary. In personal conferences with Dr. J. Fodor, Leader of the Tribological Laboratory at RIAI, the method was described. In this technique the wear debris is meticulously separated from the vehicle fluid and nuclearly activated, producing radioisotopes of the elements present in the debris. To accomplish this, the debris entrained in an oil sample is collected through a suitable filtration process and placed on the neutron flux of a nuclear reactor. After the activation process, the elemental composition of the debris is measured quanitatively using neutron activated etalons. Using computer aided techniques, the rather laborious measurements can be made rapidly. Although this technique is purported to be extremely sensitive to small amounts of wear from a critical surface, the equipment investment and the time required to perform the analysis render it completely unfeasible for the study of contaminant wear in hydraulic components.

Ferrographic wear debris analysis represents a relatively recent breakthrough in wear assessment technology. Capitalizing upon the magnetic properties [6] of the tiny wear particles, the method isolates the wear debris present in an oil sample from extraneous debris (such as ACFTD used to produce a contaminant wear situation) and the vehicle fluid. The debris thus

separated is deposited upon a specially prepared substrate where the quantity and characteristics can be measured through optical means. A Ferrographic Oil Analysis sytem was procured by Fluid Power Research Center in 1975 and has been used since that time for wear evaluation [33] [34] [35]. Based upon the results of these studies, it was felt that the Ferrographic technique represented the most practical and feasible approach to a more sensitive contaminant wear evaluation of a hydraulic component on a non-intrusive basis.

CHAPTER III

EVALUATION OF THE FERROGRAPHIC ANALYSIS METHOD

Ferrography is a technique developed to separate wear debris from a vehicle liquid and arrange it according to size on a transparent substrate for examination through optical or scanning electron microscope. Although the Ferrographic Oil Analysis system consists of several features, only those pertinent to this research effort will be discussed. Basically, the components of the Ferrographic system used consisted of two major components—a slide Ferrograph or Ferrograph analyzer, and a Ferrogram Reader. The Ferrograph Analyzer shown schematically in Figure 5 utilizes a specially developed magnet, which generates an ultra-high gradient field near its poles to force the wear particles to precipitate from the vehicle liquid. A chemically treated microscope slide is placed at an incline to the magnet and acts as a substrate upon which the wear debris is collected. A carefully controlled pressure head on the fluid is used to continually pass a low velocity stream longitudinally along the slide. After a prescribed amount of liquid has been pumped across the slide, a washing and fixing cycle removes the residual oil and causes the wear particles to adhere permanently to the slide. The amount of wear debris deposited on the slide or Ferrogram is analyzed through the use of a densitometer (Ferrogram Reader).

Due to the forces acting on the wear particles as they pass along the length of the substrate, wear debris is deposited all along the fluid path. The Ferrogram Reader, consisting essentially of a controlled light source and a photo cell, is capable of providing optical density readings at various

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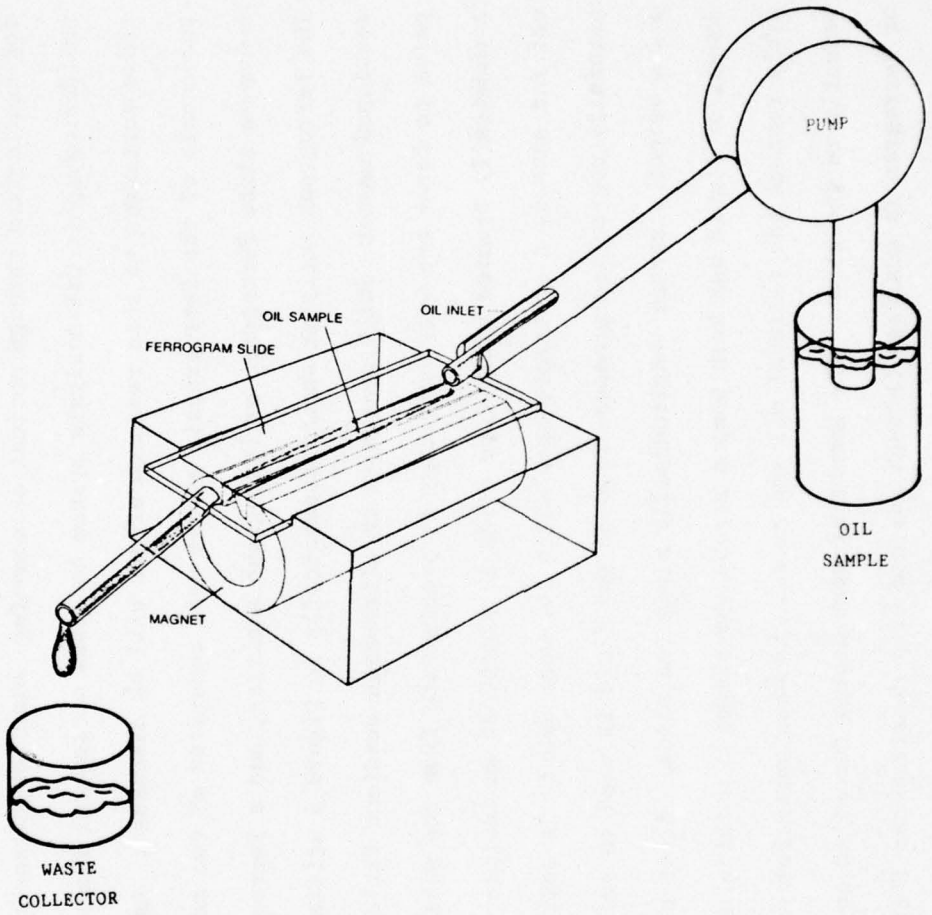


Figure 5. Schematic of Slide Ferrograph

positions along the Ferrogram. Theoretically, the density readings along the Ferrogram are a function of the particle size distribution of the wear debris since larger particles will tend to deposit sooner than smaller ones. The density readings obtained are identified by their location on the Ferrogram. For example, the sample flow is started on the Ferrogram at approximately 57 millimetres from the exit end. A density value designated as D54 indicates that it was made 54 millimetres from the exit end of the Ferrogram. In research work prior to this effort, it was found that the debris density at the 54 millimetre location was the most responsive to the contaminant wear of hydraulic pumps [34].

In 1970, Dr. W. W. Seifert, Professor of Engineering at Massachusetts Institute of Technology, and Mr. V. C. Westcott of Foxboro/Trans-Sonics, began to look for a simple method of detecting wear in machine parts [8]. Joined later by Dr. Douglas Scott of the National Engineering Laboratory at Glasgow, Scotland, this team of investigators discovered the unique magnetic properties of the wear debris normally found in fluid systems. According to Scott et al. [6], studies showed that the very small wear particles have only one magnetic domain, or at most a few domains, and hence they are among the most powerful magnets in existence. As a result of the discovery of the unusual magnetic properties of wear particles, this research team developed the instrument package that has come to be known as the Ferrographic Oil Analysis System. Since the wear particles are precipitated magnetically, virtually all of the unwanted particles (such as the AC Fine Test Dust used to produce a controlled contaminant wear process) in the fluid sample are eliminated. A more detailed description of the Ferrographic technique and associated equipment is given in Ref. 7.

Operational Procedures

There are two distinct steps in obtaining Ferrographic density readings. One step involves the preparation of the Ferrogram while the other deals with the actual optical density measurement. To prepare a Ferrogram, the fluid sample is agitated vigorously to insure uniform dispersal of the entrained debris. It has been found that heating the sample prior to agitation will enhance the dispersion process. A carefully measured volume of liquid is drawn off and placed in a clean sample bottle along with a volume of a chlorinated hydrocarbon called a fixer. The proportion of sample to fixer is normally 3 to 1. This mixture is shaken and placed on the Slide Ferrograph.

A specially cleaned and prepared substrate is placed in the slide holding fixture of the Ferrograph. The substrate is marked to insure proper placement since it has been coated with a barrier film to insure the correct channeling of the fluid stream. A delivery tube which has been accurately cut for length and has been pre-cleaned is placed into the sample bottle containing the mixture of sample fluid and fixer, positioned in a tube pump, and attached to the delivery arm of the Slide Ferrograph. The delivery arm is then lowered until the delivery tube contacts the substrate and the tube pump is energized. The tube pump is run until the sample bottle is empty to insure an exact volume of sample fluid has passed across the slide. The delivery tube is then transferred to a bottle containing only the fixer solution and the washing cycle is begun. When the correct amount of fixer has been pumped over the Ferrogram, the tube pump is stopped and the delivery arm is raised from the substrate immediately. The Ferrogram

is thoroughly dried with the aid of a high intensity lamp before it is removed from the slide holding fixture.

The finished Ferrogram is evaluated by placing it on the stage of a special microscope. The reflected light source is switched to a controlled D.C. power supply while the photo cell is aligned with the microscope barrel. Using the stage of the microscope, the reader is placed at the 55mm position and the Ferrogram is transversed perpendicular to the longitudinal axis to find the highest density. This procedure is repeated at 54.5, 54, 53.5, and 53 millimetres. These readings then are averaged to obtain the optical density value designated as D54. This averaging technique compensates for any variation in the positioning of the delivery tube or the Ferrogram on the microscope stage.

Experimental Evaluation

With the Ferrographic method, the amount of wear debris is measured directly and these measurements are subsequently used to evaluate the wear process. Hence, it is necessary to evaluate the capabilities and limitations of the Ferrographic technique to accurately measure the amount of wear debris in a fluid sample before using these data to assess a wearing process. In order to accomplish this evaluation, a sample was selected from the many pump tests which have been conducted. Various volumes of liquid from this sample were used in preparing Ferrograms to assess the repeatability and saturation characteristics of the method. The fluid volumes selected were 3, 6, 9, and 12 millilitres, and nine Ferrograms were made using each of the selected volumes. This experiment provided information relative to three

characteristics which are fundamental to the use of the Ferrographic technique. First of all, the density readings must be repeatable, that is, repeated measurement at the same debris concentration must produce consistent results. Second, since various volumes of sample liquid are used in preparing a suitable Ferrogram, it must be shown that the measurements obtained are linear with volume. This linearity characteristic would permit normalization of the data (e.g., per millilitre). Finally, the saturation tendency of the technique must be known. Figure 6 shows the results of the evaluation of the Ferrographic method using a fluid sample with debris concentration designated as A. From this figure it can be seen that if the D54 density is averaged over the nine Ferrograms made at each volume level, the results are sufficiiently linear up to a density value of between 35 and 40. The curved portion of Figure 6 indicates a saturation tendency at close to an optical density of 50 and the decision was made at this point to always obtain a value of less than 40 by proper dilution techniques. In addition, the data scatter observed from these results was greater than desired. The coefficient of variation (standard deviation divided by the mean) for the means of 10.93, 21.69, and 33.29 were 17.89%, 13.64%, and 13.48%, respectively. The 12 millilitre tests had a mean of 38.24 and a coefficient of varation of 6.45%. However, since this is definitely a saturated condition, it is not useful in evaluating the variability of the density reading. The increase in the coefficient of variation of the sample data with a mean of 10.93 indicated that the percent variation increased at low mean values. Therefore, the density values considered acceptable were above a reading of 10. Thus, if the optical density (D54) was below 10 using a given quantity of fluid, the volume was increased until the actual Ferrogram reading at 54mm exceeded 10 but was below 40.

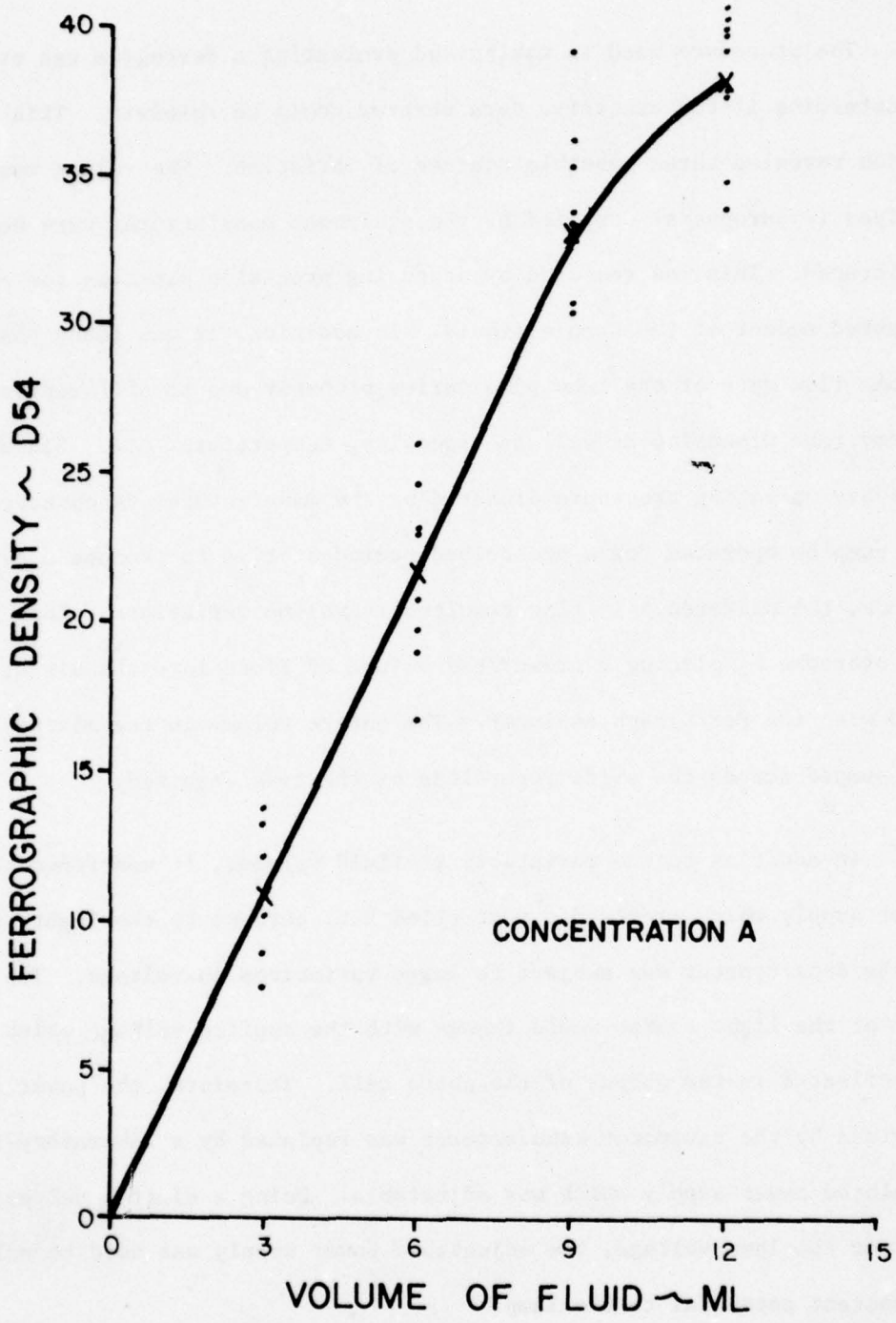


Figure 6. Repeatability and Saturation Characteristics of Ferrographic Method-Initial Procedure.

The procedure used in making and evaluating a Ferrogram was studied to determine if the excessive data scatter could be resolved. This investigation revealed three possible sources of variation. The volume measuring devices (eyedroppers) supplied by the equipment manufacturer were not accurately calibrated. This was remedied by procuring precision pipettes for extracting measured amount of the sample liquid. In addition, it was found that the volume flow rate of the tube pump varied probably due to differences in delivery tube dimension as well as viscosity, temperature, etc. Since the preliminary operating procedure dictated by the manufacturer recommended that the pump be operated for a prescribed period of time to produce a certain volume, the difference in flow resulted in volume variations. This problem was overcome by placing a prescribed volume of fluid into the mixing bottle used with the Ferrograph Analyzer. The entire volume in the mixing bottle was pumped across the slide regardless of the time required.

In addition to the variations in fluid volumes, it was found that the power supply which provided a controlled D.C. current to the light source of the densitometer was subject to large variations in voltage. The intensity of the light course would change with the applied voltage which would be reflected in the output of the photo cell. Therefore, the power supply provided by the equipment manufacturer was replaced by a laboratory-type regulated power supply which was adjustable. Using a digital voltmeter to monitor the lamp voltage, the adjustable power supply was used to maintain a constant potential to the lamp.

To assess the improvement brought about by the changes discussed, the experiment was conducted again using a sample with a different concentration

of wear debris (Concentration B). The results of this experimental evaluation of the modified Ferrographic technique is shown in Figure 7. It can be noted from the figure that the data scatter has been reduced while the linearity and saturation characteristics are approximately the same. In this experiment a sample which contained a heavier concentration of wear debris was utilized and three Ferrograms were made using liquid volumes of 0.25, 0.50, 0.75, 1, 2, and 3 millilitres. The reading obtained at 2 and 3 millilitres were far beyond saturation limits. The mean D54 reading obtained were 13.07, 24.3, 33.13, 39.2, 62.6, and 80.5 in order of increasing volumes. The coefficient of variation for these mean D54 values (using the range to estimate the standard deviation were 7.69%, 4.86%, 4.82%, 6.78%, 1.98%, and 0.62%. From these data, it was concluded that the data scatter due to the Ferrographic process had been reduced by about 50%.

Further appraisal of the Ferrographic technique led to the conclusion that additional improvements in the process would require some redesign of the hardware. For example, it is felt that the variations in the flow across the slide will result in different density readings. Since the liquid stream is contained on either side by the special boundary film, any change in flow will be reflected by an increase or decrease in the height of the stream. Thus, wear particles initially situated at the top of the fluid stream would have a different distance of travel before encountering the substrate. In addition, at higher flows the drag forces on each particle would be increased and therefore the particle would be located further along the substrate. However, in order to make the flow rate adjustable, a new feed mechanism would have to be developed and verified. Such an endeavor was considered beyond the scope of this effort.

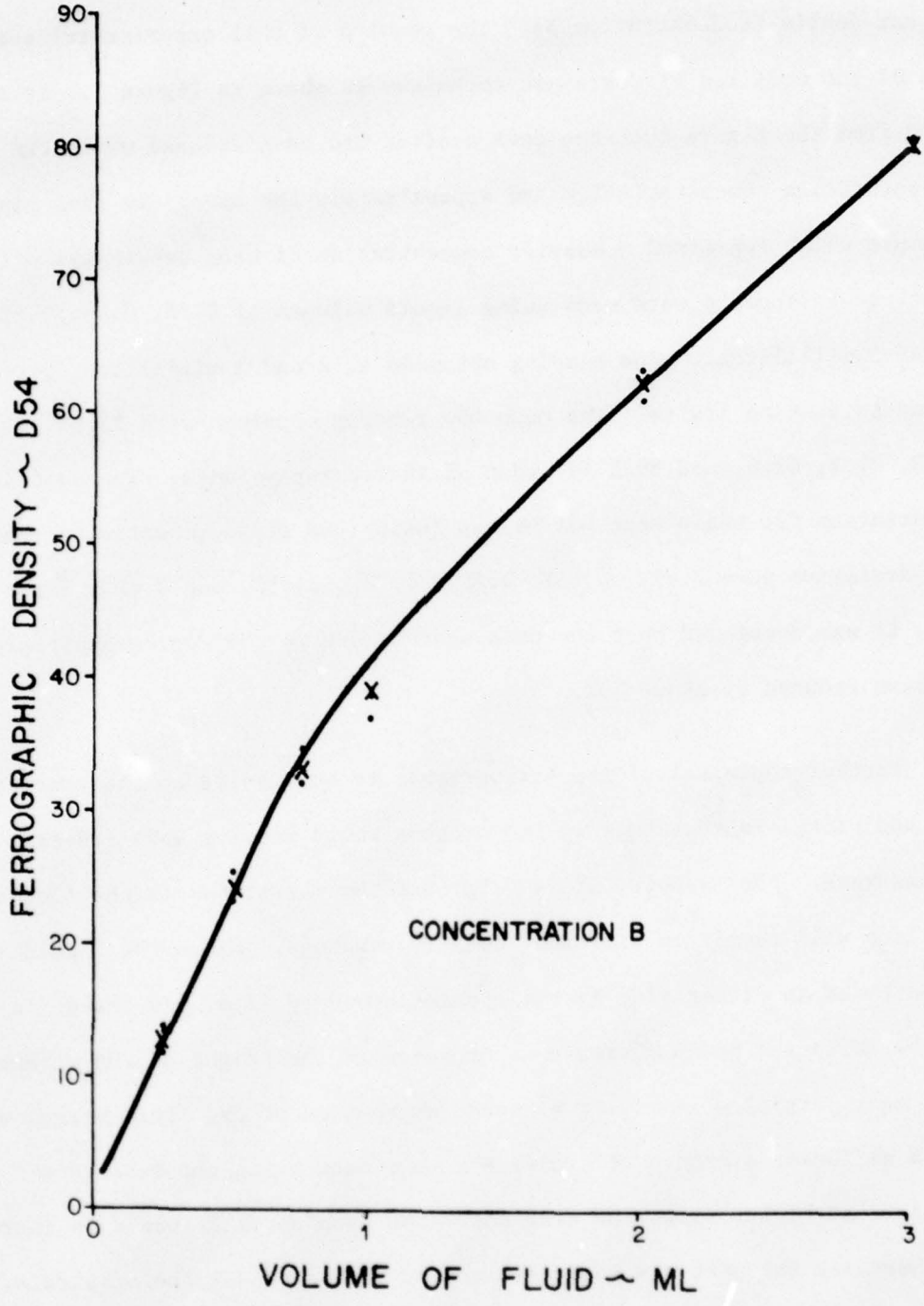


Figure 7. Repeatability and Saturation Characteristics of Ferrographic Technique-Modified Procedure.

In order to utilize the Ferrographic density readings in the evaluation of contaminant wear in a gear pump, it was necessary to develop a wear model. From the hypothesis that the primary effect of the contaminant wear process is to increase the clearance within the pump, relationship must be derived which relate Ferrographic density to the overall clearance change must be expressed in terms of flow degradation. The rationale and assumptions used in the development of a Ferrographic Wear Model for a gear pump is discussed in the next chapter.

CHAPTER IV

ANALYSIS OF CONTAMINANT WEAR USING FERROGRAPHIC DATA

The role of a pump in a hydraulic system is to deliver liquid to the other system components at a rate proportional to the speed of the pump. When this capability is no longer satisfactory, the pump is said to be worn out. As shown in the flow chart of Figure 8, when a pump operating in a system at some set of conditions (pressure, speed, temperature, etc.) is exposed to entrained contamination, the critical surfaces within the pump are abraded. The destruction of these surfaces will result in the enlargement of clearances within the pump and the generation of wear debris. Through the use of the Ferrographic technique, the amount of wear debris generated during a given contaminant exposure can be measured; however, the associated effective clearance change is impossible to measure in modern fluid power pumps.

The primary result of the increase in clearance (increased leakage of flow degradation) can be measured. The contaminant sensitivity of a gear pump could be evaluated from either flow degradation or wear debris data. However, the function of a hydraulic pump is not to generate wear debris but to produce flow. Therefore, to be ultimately useful, it is necessary to evaluate the degree of contaminant wear in a hydraulic pump in terms of flow degradation. The problem which exists in evaluating the contaminant sensitivity of a pump directly from flow loss measurements lies in the magnitude of this parameter resulting from low contamination. That is, in

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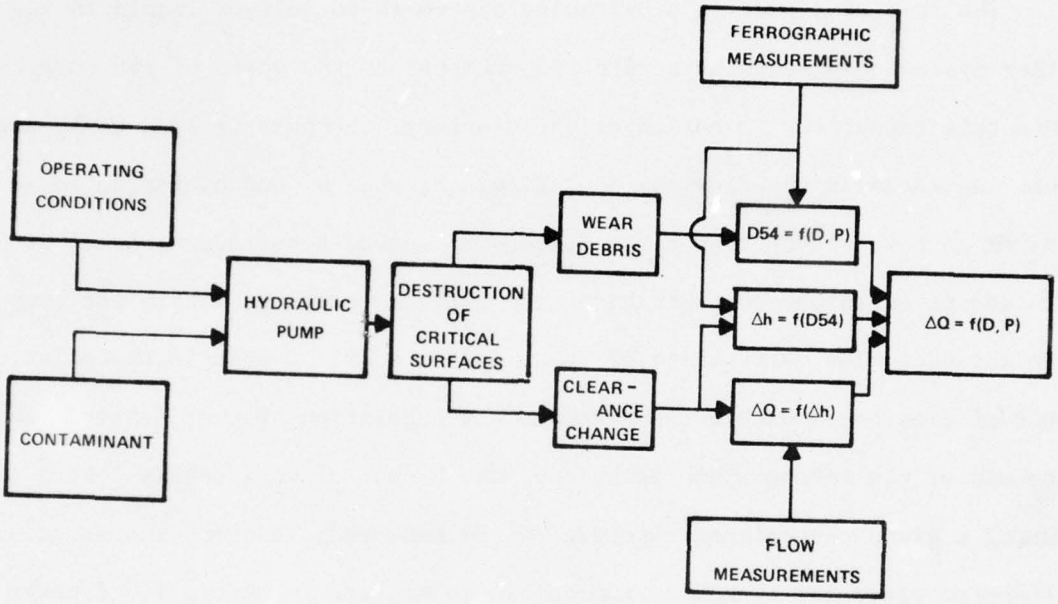


Figure 8. Flow Chart for Contaminant Wear in Hydraulic Pumps

order to produce adequate flow degradation for accurate measurement it is necessary to accelerate the contaminant wear process by using an excessive contaminant concentration (300 mg/ℓ in the standard test).

As discussed in Chapter II, previous investigations have evaluated the contaminant wear in a pump through the use of only flow degradation. This work has led to the development of a log-normal model as shown in Figure 4 for the performance degradation (flow loss) of hydraulic pumps. While this type of analysis produces useful data for interpretation of pump contaminant sensitivity, it requires an extremely high contaminant concentration. Reducing the contaminant concentration results in very small flow degradation which cannot be accurately measured.

Wear debris measurements through Ferrographic analysis offers a new method for assessing the contaminant sensitivity of a hydraulic pump. A functional relationship was presented in Chapter II connecting the amount of wear debris generated with the particle size distribution of the entrained contaminant exposed to a pump and the outlet pressure imposed. In order to use this relationship in expressing the contaminant wear of a pump in terms of flow degradation, two more expressions are needed. One of these equations can be developed by modeling the leakage paths of a pump on a lumped basis where the changes in clearances within the pump are represented by an increase in an equivalent clearance. The other expression is a coupling equation which relates the wear debris generation in terms of the change in this equivalent clearance.

The remainder of this Chapter will analyze the contaminant wear process in a gear pump to develop the form of the necessary wear model. The

Ferrographic measurement and flow evaluation during a pump contaminant wear test are independent and thus can both be used in identifying the parameters of a wear model. The next section of this chapter will investigate the flow degradation relationships which governs the contaminant wear process. Then a section will be devoted to the expressions which describe the generation of wear debris. The last section of this chapter will consider the coupling relationship which completes the Ferrographic Wear Model. From Figure 8 the Ferrographic Wear Model can be expressed in a general form as follows:

$$D54 = f_1(D,P)$$

$$\Delta h = f_2(D54)$$

$$\Delta Q = f_3(\Delta h)$$

where:

D54 - Ferrographic density measured at the 54mm position

D = particle size

P = pump outlet pressure

ΔQ = pump incremental flow loss

Δh = equivalent clearance change

Flow Degradation Due to Contaminant Wear

In general, there are three critical leakage paths inherent to hydraulic gear pumps [23], [36], [37]. One path is defined by the clearance between the tip of the gear teeth and the internal wall of the housing. A second path is across the side face of the gears to the pump bearings. Flow between this latter surface and the loading plate of the pump is used to lubricate the bearing and is relieved back to the suction of the pump. The third

path is also defined by the gear faces and the loading plate but is located at the meshing area of the gears and permits the leakage to flow directly back to the suction cavity of the pump. It should be noted that each of these leakage paths is subjected to a pressure differential and each is characterized by one moving boundary. Since these leakage paths are complex and act together to form the total leakage flow of the pump, rigorous modeling was not attempted. The objective in the following discussion is to obtain a simplified model form which is guided by experimental data from the pump.

The general expression normally used [36] [37] for describing the output flow of a gear pump can be written as follows:

$$Q_o = f(D_P, h, \Delta P, N, \mu, T)$$

where:

Q_P = output flow of pump

D_P = pump displacement

h = characteristic clearance

ΔP = pressure drop

N = speed

μ = viscosity

T = temperature

In order to determine a more explicit form of Equation 3, five gear pumps were evaluated at various pressures and speed while maintaining viscosity and temperature constant. Since the results on all five pumps (given in Appendix A) were quite similar, the data of only two of these tests, as shown in Figure 9 and 10 will be used. The five pumps tested were randomly

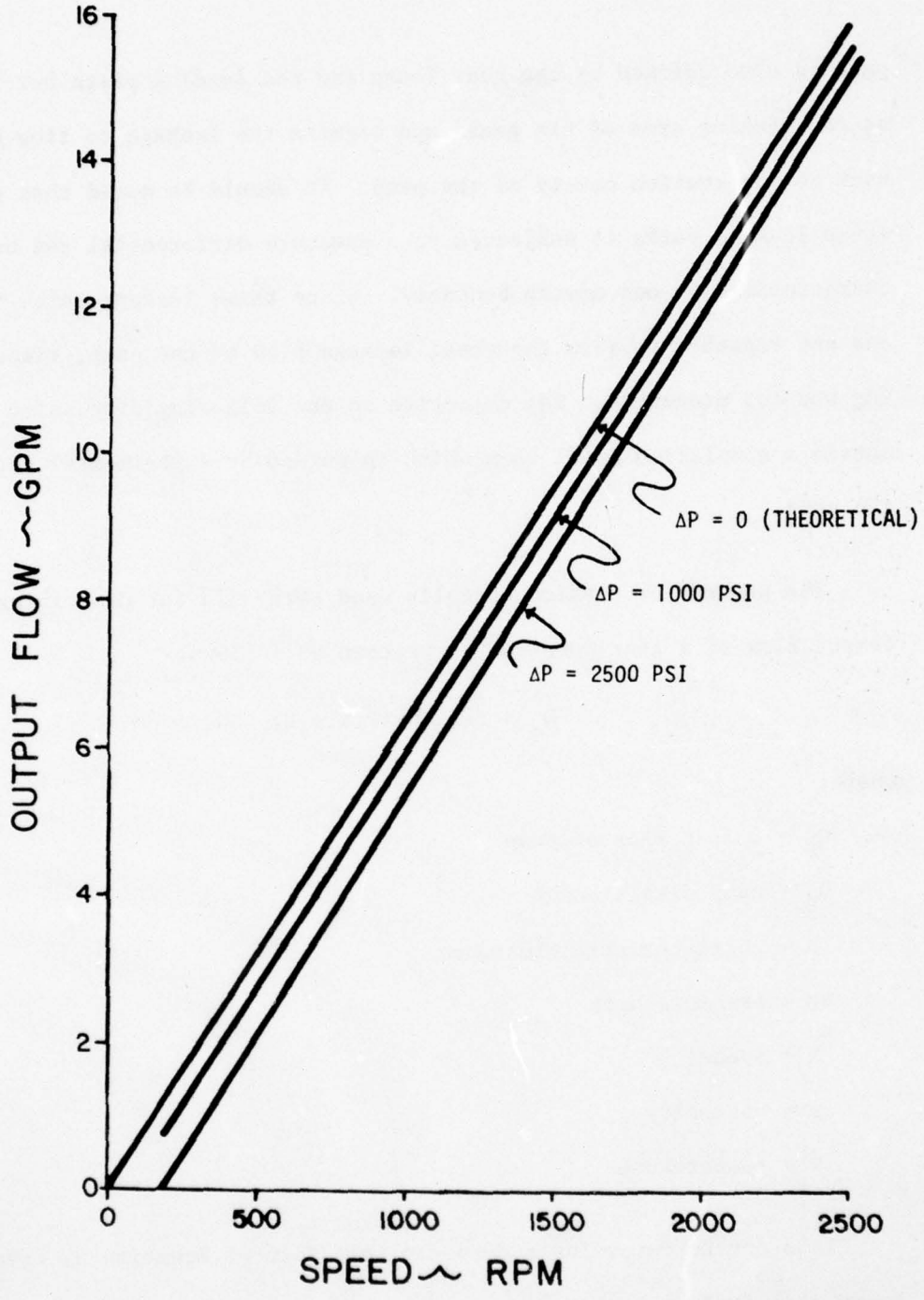


Figure 9. Results of Performance Test on Pump 302

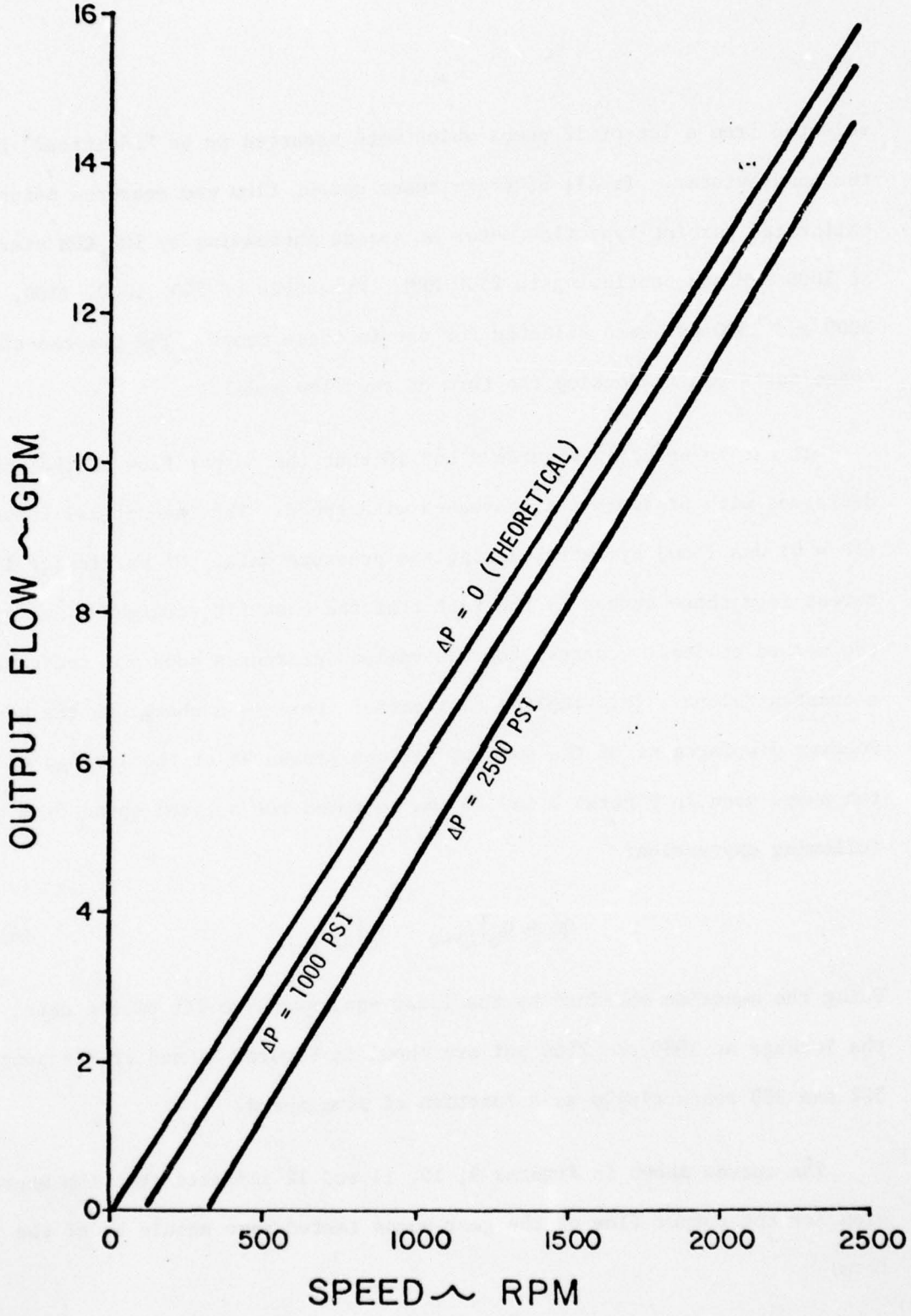


Figure 10. Results of Performance Test on Pump 303

selected from a lot of 12 pumps which were reported to be "identical" by the manufacturer. In all of these tests output flow was measured using a calibrated turbine-type flow meter at speeds increasing by 100 RPM starting at 1000 RPM and continuing to 2500 RPM. Pressures of 500, 1000, 1500, 2000 and 2500 psi were selected for use in these tests. The purpose of these tests was to develop the form of the flow model.

It can be noted in Figures 9 and 10 that the output flow of the pump decreases with pressure and increases with speed. The theoretical curve ($\Delta P = 0$) was found by extrapolating the pressure data. Of particular interest from these curves is the fact that the best fit straight line (by the method of least squares) for the various pressures does not indicate a constant slope. This implies that either there is a change in the effective displacement of the pump at various pressures or the leakage of the pumps used in Figures 9 and 10 can be found for a given speed from the following expression:

$$Q_\ell = Q_o\big|_{\Delta P=0} - Q_o\big|_{\Delta P=P} \tag{4}$$

Using the equation obtained by the least squares curve fit of the data, the leakage at 1000 and 2500 psi are shown in Figures 11 and 12 for pumps 302 and 303 respectively as a function of pump speed.

The curves shown in Figures 9, 10, 11 and 12 indicate that the expression for the output flow of the gear pumps tested here should be of the form:

$$Q_o = Q_t - Q_\ell \tag{5}$$

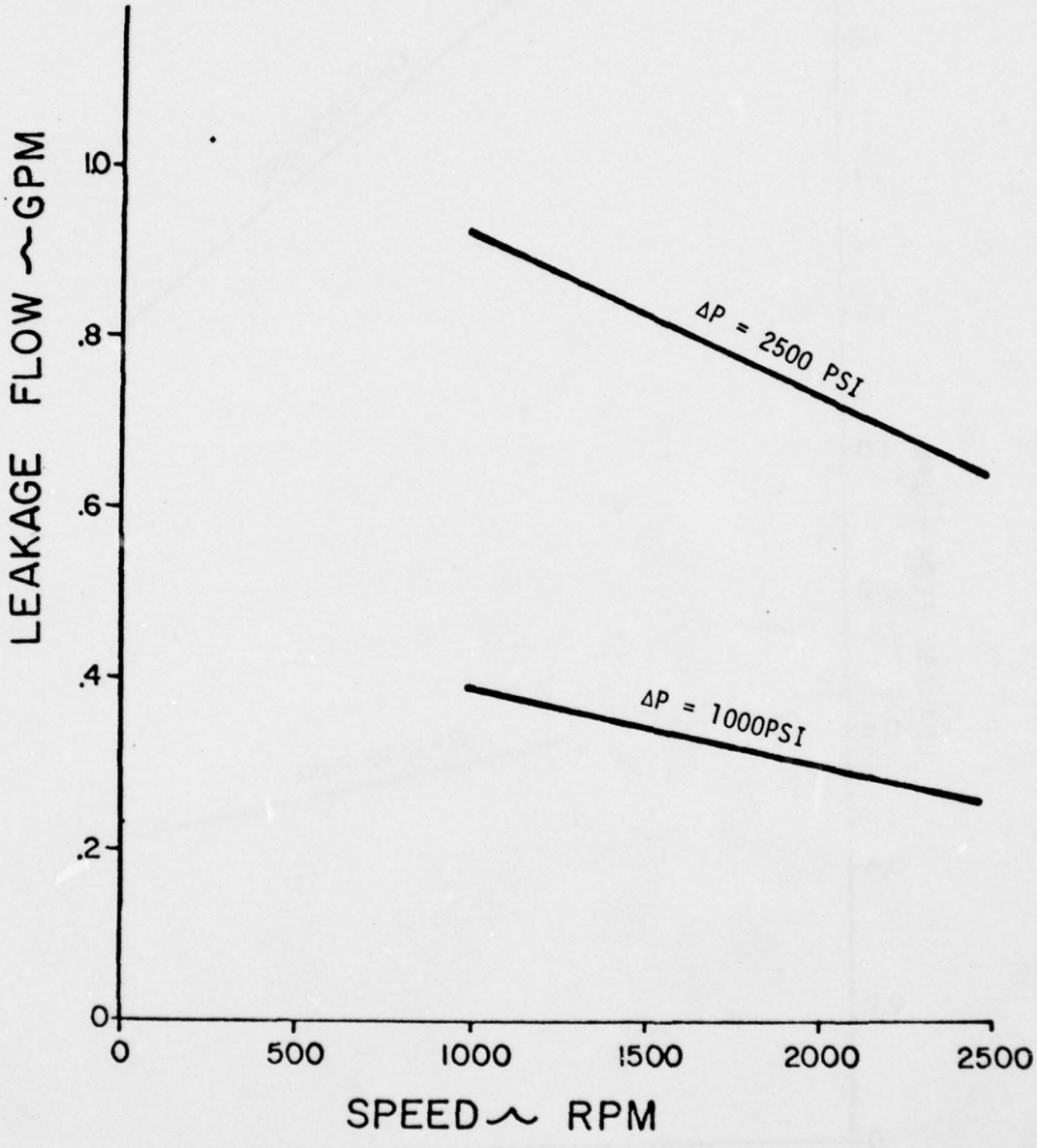


Figure 11. Leakage Flow as a Function of Speed at Various Pressures for Pump 302

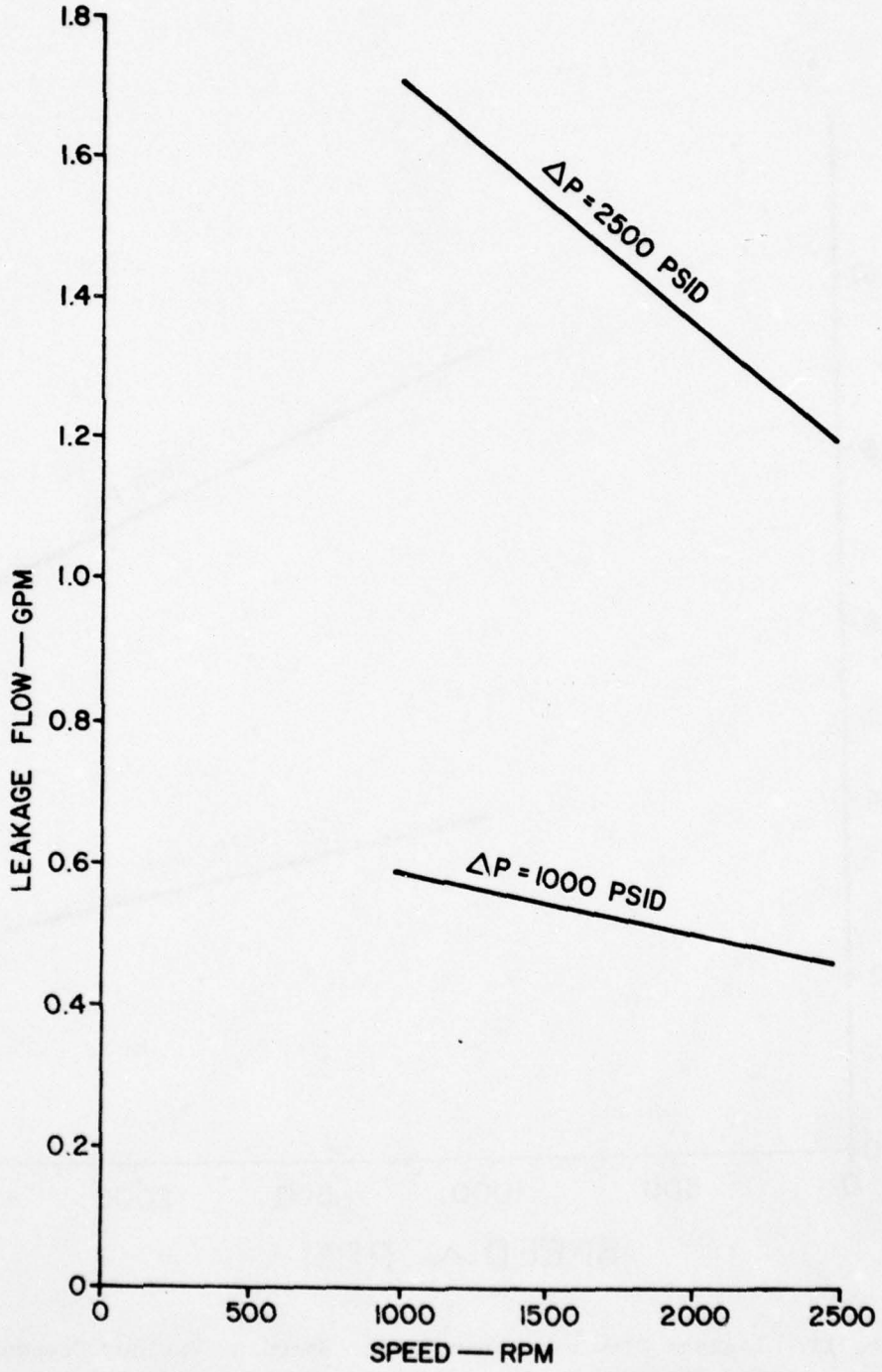


Figure 12. Leakage Flow as a Function of Speed at Various Pressures for Pump 303

where:

Q_t = theoretical flow

Q_ℓ = leakage flow

The theoretical flow, Q_t, is defined as the output flow with a zero pressure differential across the pump and can be written as follows:

$$Q_t = KD_PN \tag{6}$$

where:

K = proportionality constant

D_P = displacement found at zero pressure differential

The leakage flow, Q_ℓ, is obviously a function of both speed and pressure. Furthermore, this quantity is a linear function of these two parameters. Therefore, the expression for the leakage flow can be written as follows:

$$Q_\ell = K_1\Delta P - K_2N \tag{7}$$

where:

K_1 and K_2 = constants of proportionality

As might be expected, the form of Equation 7 suggests that flow in the leakage paths of gear pump adheres to the hydrodynamic lubrication theory for viscous flow between parallel surfaces, one fixed and one moving [36] [37]. This theory states that K_1 is a function of the clearance between the surfaces to the third power and other parameters such as length and width of the flow passage and the viscosity of the flowing liquid. In addition, the theory states that K_2 is proportional to the clearance divided by 2 which accounts for a linear velocity distribution across the flow passage. Since the parallel plate arrangement displays the same flow behavior as the

pump model depicted in Equation (7), the distance between the two parallel surfaces can be treated as the equivalent clearance of the pump.

Since the objective of this modeling exercise is to formulate a model which can be used to relate flow degradation to the equivalent clearance changes, all other variables effecting leakage flow need not be explicitly depicted. Consequently, the leakage expression, Equation (7), can be re-written to explicitly show the dependence of leakage flow upon the equivalent clearance as follows:

$$Q_\ell = K_3 h^3 \Delta P - K_4 h N \tag{8}$$

where:

K_3 and K_4 = constants of the pump

h = equivalent clearance representing the composite leakage paths

Substituting Equations (6) and (8) into Equation (5) and rearranging yields the following relationship.

$$Q_o = KD_P N + K_4 h N - K_3 h^3 \Delta P \tag{9}$$

Equation (9) is sufficiently well defined for the development of an expression for use in contaminat wear evaluation, since such wear tests are conducted at constant speeds and pressures.

In a contaminant sensitivity test the pump is subjected to a given size range and concentration of AC Fine Test Dust. Damage to the pump results in an increase in the effective clearance represented by h in Equation (9). Therefore, the flow of the pump after the first exposure can be written as follows:

$$Q_{of} = KD_P N + K_4 (h+\Delta h) N - K_3 (h+\Delta h)^3 \Delta P \tag{10}$$

where:

Q_{of} = output flow after contaminant exposure

Δh = change in the equivalent clearance of the pump due to contaminant wear

The flow degradation, therefore, can be found by subtracting Equation (10) from Equation (9). Performing this manipulation and simplifying produces the following relationship:

$$\Delta Q = (3K_3h^2\Delta P - K_4N)\Delta h + (3K_3h\Delta P)\Delta h^2 + (K_3\Delta P)\Delta h^3 \quad (11)$$

where:

ΔQ = flow degradation due to contaminant wear

Since it would be impossible to quantify each of the variables of Equation (11), for a given pressure drop, speed, and initial value of clearance, it can be further reduced as follows:

$$\Delta Q = C_1\Delta h + C_2\Delta h^2 + C_3\Delta h^3 \quad (12)$$

The expression given by Equation (12) represents the transformation needed to relate the wear caused by contamination (i.e., Δh) to the resulting flow degradation. Unfortunately, there is no way of nonintrusively measuring the independent variable, Δh, of this relationship directly. Therefore, this variable must be evaluated indirectly which is the purpose of the wear debris analysis technique.

Debris Generation From a Gear Pump

Any wear process which results in a destruction of a material surface produces debris. If during the wear of a flow path only one dimension is altered, then the amount of wear debris generated will be directly proportional

to the dimensional change of the path. It is extremely difficult to visualize and illustrate the leakage paths of a gear pump although they can be adequately described. For descriptive purposes it will be assumed that each of the three critical leakage paths are rectangular.

Consider, first, the leakage path defined by the sides of the gears and the surface of the loading or wear plate as it is commonly called. Fluid which enters this space must either flow to the bearing cavities or across the meshing area of the gears directly to suction. The liquid flow to the bearing cavities is indicated as Q_1 in Figure 13 which shows the cross-section of a gear pump in schematic form. The leakage which crosses the meshing area of the teeth is called Q_2 in Figure 13. The length of the path in which Q_1 flows is determined by the root diameter of the gear teeth and the hole in the wear plate through which the pump shaft passes. The width of the Q_1 path is the mean arc length between the root diameter of the gear and the diameter of the shaft. On the other hand, the length and width of the path through the gear mesh area are controlled by the length and width of the gear teeth. In neither of these two paths is contaminant wear likely to alter the length or width of the leakage paths to any appreciable degree. The clearance of both paths is the distance between the gear faces and the surface of the wear plate and this is the dimension which is altered during contaminant wear. It is logical to conclude, therefore, that material removed from the gear faces or wear plate surface will be directly reflected in the clearance change between them.

The leakage path defined by the tip of the gear teeth and the body of the pump has a length proportional to the distance across the gear tip.

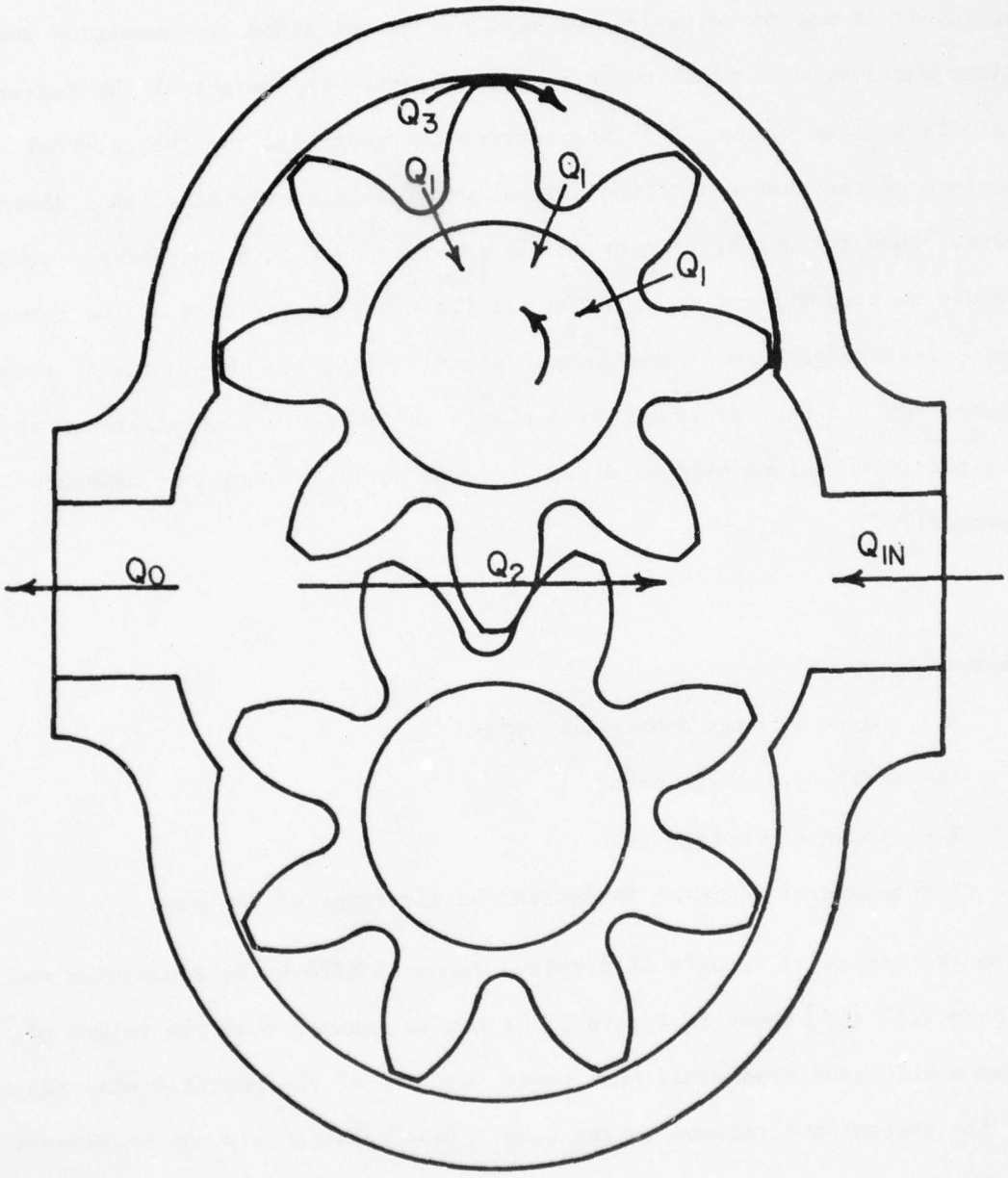


Figure 13. Schematic Cross-Section of a Gear Pump

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The width of this leakage path is equal to the width of the gear. Here again, it is not conceivable that wear due to entrained contamination would alter the length or width of this leakage path. The height of the leakage path is defined by the clearance between the gear tips and the internal surface of the pump body. Contaminant wear could easily alter this dimension. Thus the amount of wear debris generated due to a contaminant attack should be proportional to only the clearance change for each of the three major leakage paths in a gear pump. Assuming that the net change in these clearances can be represented by a change in the equivalent clearance of the pump, the volume of wear debris removed during contaminant exposure becomes:

$$V = wl\Delta h \tag{13}$$

where:

V = volume of wear debris generated

w = width of leakage path

L - length of leakage path

Δh = change or increase in equivalent clearance of the pump

From the published results of previous research efforts by Rabinowicz and others [15] [17] shown in Figure 1, it can be assumed that the volume of wear debris generated would be a power function of the particle size range of the contaminant exposed to the pump. This relationship can be expressed as follows:

$$V = aD^b \tag{14}$$

where:

a = constant coefficient

D = particle size of the contaminant

b = constant proportional to the loading conditions which exist between the two wearing surfaces

Equation (14) and Equation (13) can be combined to produce the following expression for the flow degradation in terms of the particle size distribution and concentration of the contaminant to which the pump was exposed as follows:

$$\Delta Q = C_1 \left(\frac{a}{wL}\right) D^b + C_2 \left(\frac{a}{wL}\right)^2 D^{2b} + C_3 \left(\frac{a}{wL}\right)^3 D^{3b} \qquad (16)$$

To utilize this equation, the constants must be identified from experimental data. If a low contaminant concentration is used (below 300 mg/ℓ) there would be very little flow degradation making the identification procedure difficult and of dubious accuracy. However, by measuring the amount of wear debris through Ferrographic techniques the constants a/wL and be can be evaluated leaving only C_1, C_2, and C_3 to rely upon flow degradation data.

Ferrographic Density and Debris Concentration

The Ferrographic technique is actually a means of separating the wear debris present in an oil sample from both the vehicle fluid and extraneous material. From the results reported during the development and verification of the technique (documented in the various references) it is logical to conclude that the amount of debris deposited upon the Ferrogram would be proportional to the concentration of the wear debris in the sample fluid. Furthermore, research work conducted by the author has revealed that the

optical density measured at the 54 mm location on the Ferrogram (D54) is the most responsive to the contaminant wear debris. This leads to the development of the following expression.

$$V = C_4(D54) \tag{17}$$

where:

C_4 = a constant of the Ferrographic process

D54 = optical density measured at the 54 mm location on a Ferrogram

By substituting Equation (17) into Equation (14) the following expression is produced.

$$D54 = \frac{a}{C_4} D^b \tag{18}$$

Since various amounts of sample fluid can be used in obtaining the density value, even a very slight amount of debris can be accurately assessed. That is, if the contaminant wear process produced a flow change which was too small to measure practically, a very small amount of wear debris would be generated. However, by passing a large amount of sample fluid across the substrate, a sufficient quantity of wear debris will be deposited to permit accurate evaluation if the limitations of the Ferrographic technique are adequately respected.

Ferrographic Wear Model

From the preceding development, an expression for the flow degradation in terms of the equivalent clearance change, Equation (12), and a relationship for the Ferrographic density as a function of contaminant exposure, Equation (18), have been advanced. However, as they now stand

these two equations are independent and, in fact, the equivalent clearance change used in Equation (12) cannot be measured. Therefore, to complete the Ferrographic Wear Model, an expression must be found which couples the two describing equations. By examining Equations (13) and (17), it can be seen that both the equivalent clearance change, Δh, and the Ferrographic density, D54, are linearly related to the volume of debris generated. This relationship can be expressed as follows:

$$\Delta h = K_5(D54) \tag{19}$$

The Ferrographic Wear Model, then, consists of Equations (18), (19), and (12) and can be written as follows:

$$D54 = a_1 D^b$$

$$\Delta h = K_5(D54) \tag{20}$$

$$\Delta Q = C_1\Delta h + C_2\Delta h^2 + C_3\Delta h^3$$

It whould be noted that the three expressions comprising the Ferrographic Wear Model could be combined to produce only one equation. However, the results of such a combination would obscure the true meaning of the model and preclude the use of both measurements, (D54 and ΔQ) in identifying the individual parameters of the Ferrographic Wear Model.

To use this model, a contaminant sensitivity test would need to be conducted on the gear pump. A fluid sample would be extracted at the conclusion of each contaminant exposure. However, the contaminant concentration could be greatly reduced from the 300 mg/ℓ presently used since the Ferrographic analysis is much more sensitive to small changes in clearance than is flow changes. Since a density reading will be

obtained at each contaminant exposure regardless of whether or not the flow change is measurable, several data points can be obtained to identify the constants of the first equation in the model without excessive degradation in flow. The successive exposures (0-5, 0-10, 0-20, etc.) need only be continued until a measurable flow degradation has occurred after two exposures. This will produce sufficient data for the identification of the constant in the last expression of the model. Since very little damage has been done to the pump during this test, a new test can be conducted on the same pump operating at a different set of operating conditions (e.g., a different pressure).

An extensive testing program was conducted to verify the relationships which have been developed. In all 24, pump tests were conducted, not counting the experiments to evaluate the performance characteristics of the the test pumps. These tests were run at various contaminant concentrations, pressure, speed, and sequence. The sequence tests were made to determine the significance of running a gear pump more than once to evaluate the contaminant wear characteristics at different operating conditions. The results of these tests are presented in the following chapter.

CHAPTER V

EXPERIMENTAL VERIFICATION OF FERROGRAPHIC WEAR MODEL

There are numerous parameters of almost any hydraulic component which cannot be measured directly. For example, most hydraulic gear pumps used today are termed either pressure loaded or wear compensated. This means that such pumps include a plate which is balanced against the sides of the gears by hydrostatic and hydrodynamic forces. Therefore, it is not possible to accurately measure the clearance of any of the critical leakage paths within a gear pump. Thus, in order to verify the model developed in Chapter IV it was necessary to conduct tests in which both the flow degradation and the Ferrographic data were obtained. The degree of correlation between these two sets of data will provide the verification of the Ferrographic Wear Model.

To accomplish the testing phase of this effort, 12 gear pumps were obtained from a single manufacturer. These pumps were from the same manufacturing lot and carried identical part numbers. However, as can be seen from the performance data presented in Chapter IV and Appendix A, there was considerable variation in the initial volumetric efficiency of the pumps. This variation was not expected to cause any problem in this effort because the results are primarily concerned with change in performance (flow) and no total volumetric efficiency.

Experimental Methods

The 12 pumps were first randomly numbered commencing with 300 and concluding at 311. The test plan followed is shown in Figure 14. As can be seen from this plan, speeds of 1500, 2000, and 2500 RPM and pressures of 1500, 2000, and 2500 psi were selected. These ranges covered the normal spectrum of operating conditions used for gear pumps. The contaminant concentration levels used as shown in Figure 14 were 10, 20, 25, 75, 150 and 300 milligrams per litre. While not shown in this figure, the particle size ranges used were 0-5, 0-10, 0-20, 0-30, 0-40, 0-50, 0-60, 0-70, and 0-80 micrometres obtained by classification using air elutriation from AC Fine Test Dust [48].

The equipment utilized for conducting these tests is identical to that specified in the standard pump contaminant sensitivity test. Briefly, the test stand consisted of a 100 hp constant speed motor which drove the pumps through a belt and pulley arrangement. Various sized pulleys were used to obtain the different test speeds. The pressure at the output of the pump was measured by a calibrated bourdon type pressure gage and maintained through the use of a needle valve placed in the flowing circuit. The test circuit is shown schematically in Figure 15. Heat exchangers utilizing cool water provided adequate temperature control while temperature measurement was accomplished through the use of a gas filled monitoring system. A highly efficient filtering sub-system which could be valved into the main circuit was utilized to remove both the contaminant and wear debris after each exposure. The output flow of the test pump was monitored by a calibrated turbine type flow meter.

SPEED (RPM)	PRESSURE (PSID)	CONT. LEVEL MG/ℓ	PUMP NUMBER
1500	1500	20	307-1
	2000	20	305-2
	2500	20	307-2
2000	1500	20	300-5
		20	306-1
		20	308-2
		150	304
	2000	0	300-1
		10	300-2
		20	300-3
		20	306-2
		20	308-3
		25	302-1
		75	309
		150	301
		300	310
		300	311
	2500	20	300-4
		20	306-3
		20	308-1
		150	303
2500	1500	20	305-3
	2000	20	307-3
	2500	20	305-1

Figure 14. Experiment Test Plan

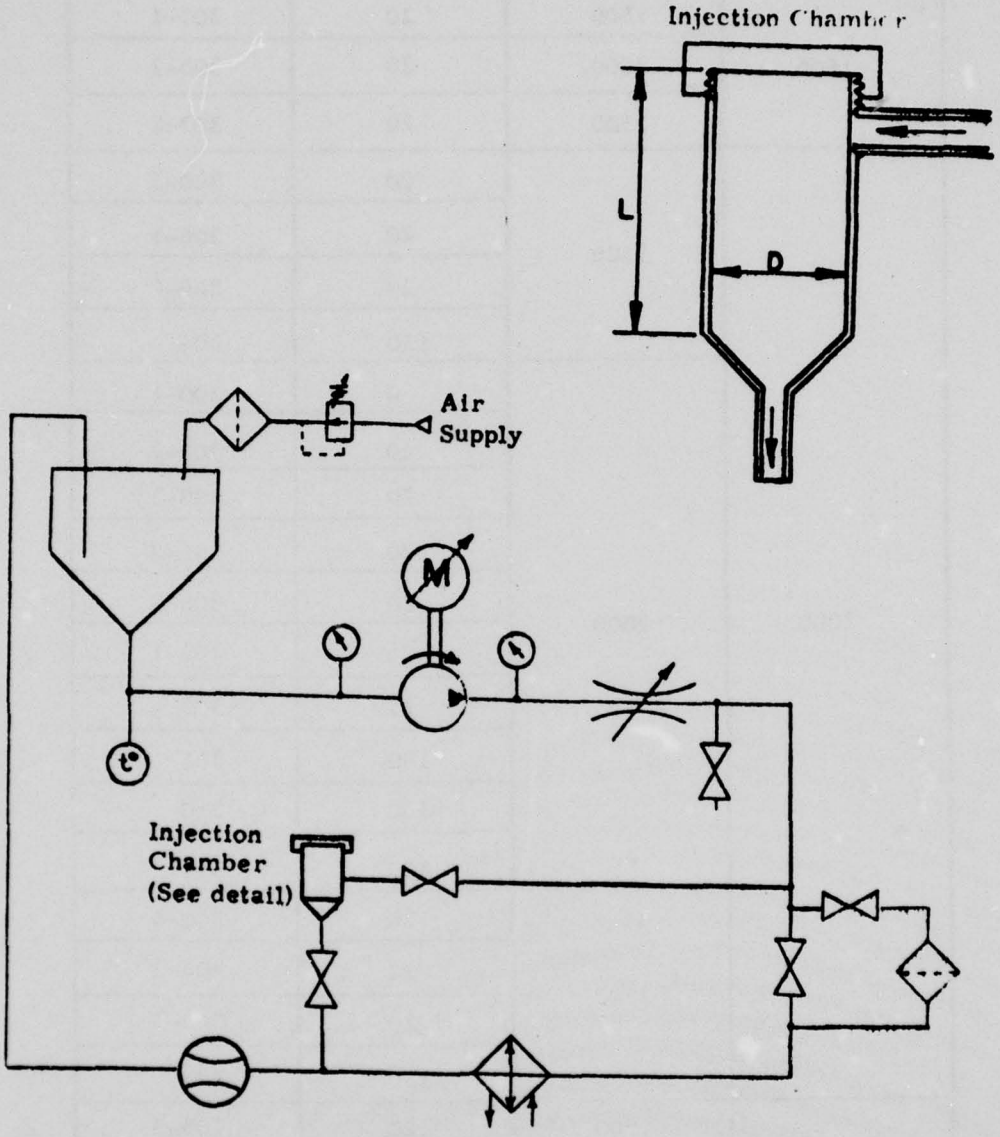


Figure 15. Schematic of Test Circuit

The test procedure followed is outlined in Appendix B. Basically, the standard contaminant sensitivity test [10] was used with the exception of the contaminant concentrations which were selected. The test pumps were all subjected to a break-in period consisting of operating the pump sequentially at 2500 RPM and 625 psid, 2500 RPM and 1250 psid, 2500 RPM and 1875 psid, and then at 2500 RPM and 2500 psid until the measured output flow remained constant for a period of one hour. Then five of the pumps were tested at various pressures and speed to evaluate the performance characteristic. The performance data is included in Appendix A. The performance tests did not damage the pumps so they were also used for contaminant wear tests.

In the wear tests the pump was operated at the desired conditions of speed and pressure until the temperature stabilized at 150°F. The initial flow rate of the pump was accurately measured and then the pump was exposed to the required concentration of contaminant in the 0-5 micrometre size range. The pump was operated with this entrained contamination level until the flow remained constant for 10 minutes or until 30 minutes had elapsed. At this point a fluid sample was extracted from the system for Ferrographic analysis and the control filters were valved into the main test system. The filtering period was 10 minutes after which an accurate flow measurement was made. The filtering sub-system was removed from the test system and the 0-10 micrometre size range was injected. This procedure was repeated for 0-20, 0-30, 0-40, 0-50, 0-60, 0-70, and 0-80 micrometre particle size ranges sequentially.

Test Results

The flow degradation data obtained during the contaminant wear tests are included in Appendix C while the Ferrographic results are shown in Appendix D. Before using the test data to evaluate the Ferrographic Wear Model, it was subjected to various statistical analyses to determine which of the various test variables had a significant influence on the test outcome. The test plan was arranged to provide a latin square statistical sub-plan which could be effectively used to evaluate the influence of conducting more than one test on a given pump. In addition, a factorially arranged sub-plan was included to be used in appraising the significance of both speed and pressure. Furthermore, a series of tests were conducted at constant speed and pressure (2000 RPM and 2000 psid) to evaluate the influence of concentration and to select the minimum level at which realistic results could be obtained. The latin square arrangement is shown in Figure 16 while the factorial arrangement is illustrated in Figure 17.

The objective in arranging the experimental tests in this manner was, of course, to evaluate the effect of these various variables, but it should be recognized that if the statistical tests reveals no significant influence from any of the variables, the data can be averaged to provide a stronger test base. For example, if the standard analysis of variance reveals that the sequence of testing has no significant effect, then the results of tests on pumps 300, 306, and 308 can be appropriately averaged to obtain a better estimate of the influence of other variables such as speed and pressure.

The statistical analysis of the latin square arranged tests revealed that the sequence of testing did not have a significant influence on the

SEQUENCE			
PRESSURE (PSID)	TEST NO 1	TEST NO 2	TEST NO 3
1500	306	308	300
2000	300	306	308
2500	308	300	306

Figure 16. Latin Square Arrangement for Test Sequence

SPEED / PRESS	1500	2000	2500
1500	307-1	300-5 306-1 308-2	305-3
2000	305-2	300-3 306-2 308-3	307-3
2500	307-2	304-4 306-3 308-1	305-1

Figure 17. Factorial Arrangement for Speed and Pressure

test results at the 5% significance level. In addition, a similar analysis of the factorially arranged tests indicated that at the 5% significance level pump speed did not have a significant effect upon contaminant wear of a gear pump. The former result (sequence makes no difference) is a necessary condition for the research to have far reaching benefits. This means that a pump can be tested at several different pressures and the results will be meaningful. The second result is not surprising since the reported work at Rabinowicz and others, discussed in Chapter II, revealed a similar conclusion. The calculations of the statistical analysis performed are shown in Appendix E. The test results can be reduced, therefore, and the speed and sequence parameter can be eliminated by averaging. Once the data was reduced, the effect of pressure was tested statistically and found to be significant for the larger particle sizes at the 5% significance level.

It should be noted that since the wear test was conducted by sequentially exposing the pump to increasing particle size ranges, the output flow was a function of the total clearance change. Thus, the change in flow or ΔQ was taken to be the difference between the output of the pump after the exposure and the flow of the pump initially. To be comparable then, the Ferrographic density data had to be accumulated. That is, the change in flow after say the 0-20 micrometre size range exposure was the result of the clearance increase due to the 0-5, 0-10, 0-20 micrometre exposures. Therefore, the total amount of wear debris generated must be proportional to the total clearance change. This can be illustrated as follows:

$$Q_{0-20} - Q_0 = \Delta Q_{0-20} \; \alpha \Delta h_{0-5} + \Delta h_{0-10} + \Delta h_{0-20} \qquad (21)$$

where:

ΔQ_{0-20} = flow degradation after the 0-20 micrometre size range exposure

Δh_{0-5} = change in clearance due to 0-5 exposure

Δh_{0-10} = change in clearance due to 0-10 exposure

Δh_{0-20} = change in clearance due to 0-20 exposure

let:

$$\Delta h_{0-5} = h_{0-5} - h_0 \tag{22}$$

where:

h_{0-5} = total clearance after 0-5 exposure

h_0 = initial clearance

and:

$$\Delta h_{0-10} = h_{0-10} - h_{0-5} \tag{23}$$

and:

$$\Delta h_{0-20} = h_{0-20} - h_{0-10} \tag{24}$$

Substituting Equations (22), (23), and (24) into Equation (21) produces

$$\Delta Q_{0-20} \alpha\ h_{0-20} - h_0 \tag{25}$$

Now let:

$$\Delta h_{0-5} \alpha\ D54_{0-5} \tag{26}$$

and:

$$\Delta h_{0-10} \alpha\ D54_{0-10} \tag{27}$$

and:

$$\Delta h_{0-20} \alpha\ D54_{0-20} \tag{28}$$

Then it follows that:

$$\Delta Q_{0-20} \propto \sum_{0}^{D} D54_{0-D} \tag{29}$$

Furthermore, it must be pointed out that in order to obtain the best possible Ferrographic density data, various fluid volumes were used depending upon the debris concentration in the fluid. Therefore, the Ferrographic densities must be normalized to a constant volume to be comparable. In this case, all Ferrographic densities have been normalized to one millilitre of sample fluid.

Since the statistical analysis revealed that neither speed nor test sequence had a significant influence upon the test results, the results of tests which were conducted to evaluate these factors were merged with the other test results to produce a reduced data set. Table I shows the flow degradation results given in terms of the flow degradation ratio which is defined as the output flow. Table II reveals the total change in flow calculated by subtracting the output flow after each exposure from the initial flow. The reduced Ferrographic density data is given in Table III where the D54 readings, normalized to one millilitre have been accumulated.

The flow degradation results can be graphically illustrated by plotting the flow degradation ratio as a function of the upper limit of the particle size range for the various concentrations used in the test. Figure 18 shows the flow degradation ratio versus particle size range for the pump tests conducted at 2000 psid pressure differential. It can be seen from this figure that the curves for the various concentrations seem to be converging.

TABLE I

AVERAGED FLOW DEGRADATION RATIO RESULTS

PRESS (PSID)	CONT LEVEL (MG/ℓ)	FLOW DEGRADATION RATIO AT INDICATED SIZE $\frac{Q_f}{Q_i}$								
		0-5	0-10	0-20	0-30	0-40	0-50	0-60	0-70	0-80
1500	20	1.0	1.0	1.0	0.999	0.996	0.991	0.986	0.982	0.971
	150	1.0	1.0	0.987	0.987	0.867	0.804	0.720	0.661	0.644
2000	10	1.0	1.0	1.0	1.0	1.0	1.0	0.996	0.994	0.987
	20	1.0	1.0	1.0	0.996	0.991	0.982	0.975	0.965	0.954
	25	1.0	1.0	1.0	0.997	0.987	0.978	0.966	0.956	0.941
	75	1.0	1.0	0.997	0.981	0.952	0.920	0.884	0.849	0.798
	150	1.0	1.0	0.990	0.944	0.885	0.842	0.765	0.712	0.653
	300	1.0	0.997	0.959	0.875	0.769	0.707	0.582		
2500	20	1.0	1.0	1.0	0.998	0.990	0.979	0.969	0.951	0.935
	150	1.0	1.0	0.992	0.947	0.876	0.796	0.717	0.655	0.636

TABLE II

AVERAGED FLOW DEGRADATION RESULTS

PRESS (PSID)	CONT LEVEL (MG/ℓ)	FLOW DEGRADATION AT INDICATED SIZE, ΔQ								
		0-5	0-10	0-20	0-30	0-40	0-50	0-60	0-70	0-80
1500	20	0	0	0	.01	.05	.10	.16	.22	.34
	150	0	0	.15	.72	1.52	2.24	3.19	3.87	4.06
2000	10	0	0	0	0	0	0	.04	.07	.15
	20	0	0	0	.05	.10	.19	.27	.39	.51
	25	0	0	0	.04	.16	.27	.42	.54	.73
	75	0	0	.04	.23	.57	.95	1.37	1.79	2.39
	150	0	0	.12	.69	1.41	1.94	2.89	3.54	4.26
	300	0	.04	.51	1.56	2.87	3.65	5.20		
2500	20	0	0	0	.02	.11	.24	.35	.55	.73
	150	0	0	.09	.62	1.46	2.41	3.34	4.07	4.30

TABLE III

AVERAGED FERROGRAPHIC DENSITY RESULTS

PRESS (PSID)	CONT LEVEL (MG/ℓ)	FERROGRAPHIC DENSITY AFTER INDECATED SIZE, Δ D54/mℓ								
		0-5	0-10	0-20	0-30	0-40	0-50	0-60	0-70	0-80
1500	20	1.13	2.4	4.37	6.57	9.10	12.04	15.69	19.61	22.8
	150	7.95	16.85	44.85	88.52	120.85	155.18	206.85	254.52	294.85
2000	10	1.11	2.52	4.33	6.04	7.53	9.24	11.44	13.27	15.22
	20	1.05	2.78	4.85	7.56	10.98	14.72	18.18	22.52	26.51
	25	1.75	4.28	8.21	11.46	16.18	19.58	24.35	28.20	34.40
	75	3.85	7.52	12.95	21.32	33.32	44.92	56.52	68.52	78.52
	150	8.40	19.0	47.67	77.17	118.34	141.17	174.47	201.3	236.63
	300	19.20	42.75	87.00	152.40	218.40	304.65	378.82		
2500	20	1.42	3.44	6.26	10.04	14.73	19.23	24.76	31.13	35.88
	150	5.0	19.8	51.8	86.8	151.47	202.8	272.47	333.14	415.14

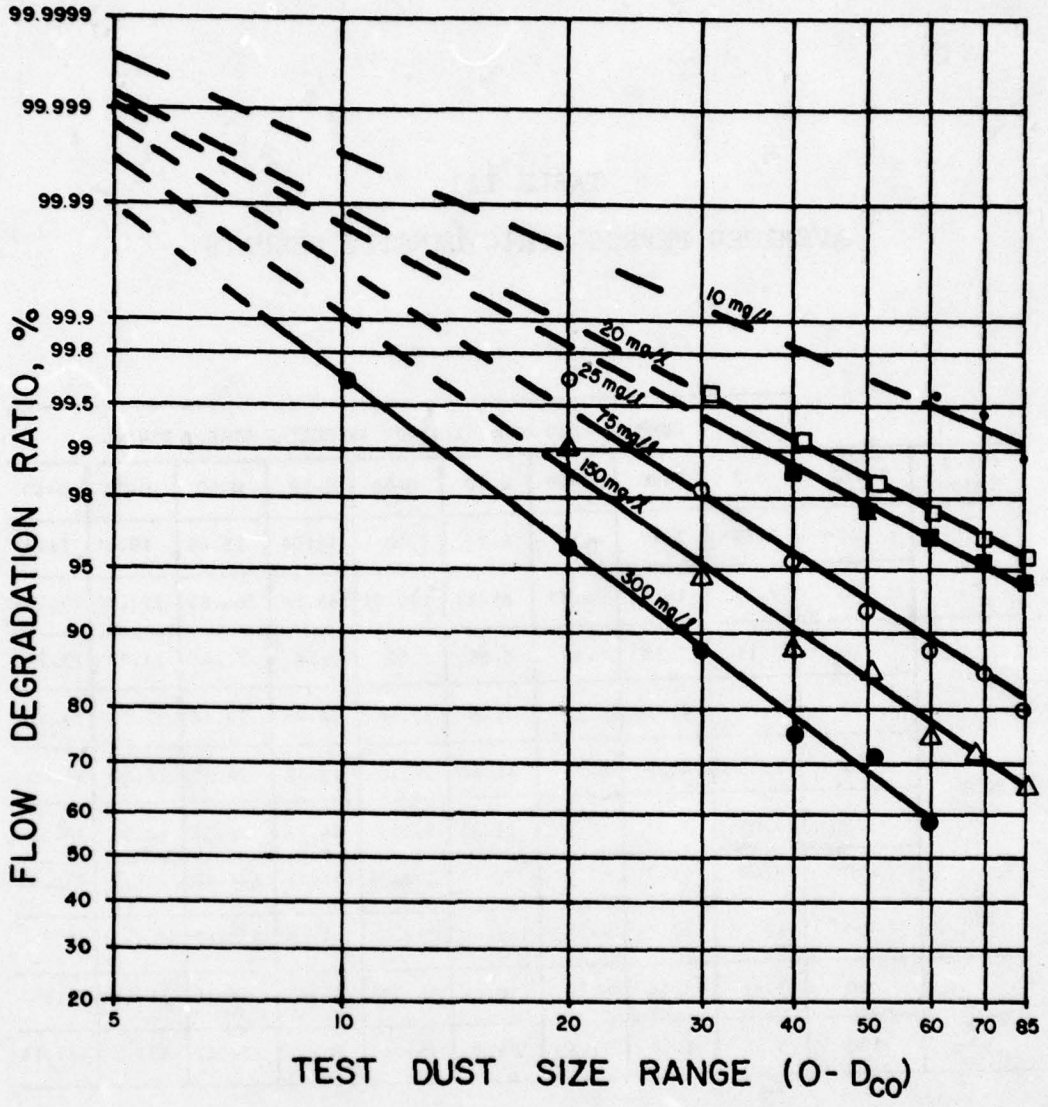


Figure 18. Flow Degradation Ratio as a Function of Particle Size Range for Various Concentrations at 2000 psid

This is not entirely surprising since at some particle the wear process would be dominantly erosive where concentration would have a much reduced effect. The flow degradation ratio versus particle size range for the various pressure level utilized and a concentration of 20 mg/ℓ is shown in Figure 19. Here again the curves seem to converge implying that below a given particle size range pressure does not have a significant influence upon the contaminant wear process for a gear pump.

The Ferrographic density data from Table III can best be graphically illustrated by plotting the Σ D54/ml as a function of the particle size range exposed to the gear pump. Such curves have been plotted for various concentrations in Figure 20 and for various pressures in Figure 21. The curves shown in these two figures are least squares best fit of a power function to the data shown. It can be seen from these curves that concentration at a given pressure tends to move the curve upward in a parallel fashion. On the other hand, pressure at a given concentration causes the slope of the curves to be altered. Considering the data shown in Figures 18 and 20, it can be seen why 20 mg/ℓ was selected for the bulk of the tests. At this concentration, it was felt that there was reasonable flow degradation and wear debris generation to produce a meaningful correlation. At 25mg/ℓ, the flow degradation was considered excessive and at 10 mg/ℓ, the wear debris generation as measured Ferrographically was low.

Verification of Gerrographic Wear Model

The Ferrographic Wear Model (equation (20)) as presented in Chapter IV stated that the Ferrographic density was a power function of the particle size range exposed to the gear pump. This part of the model was verified

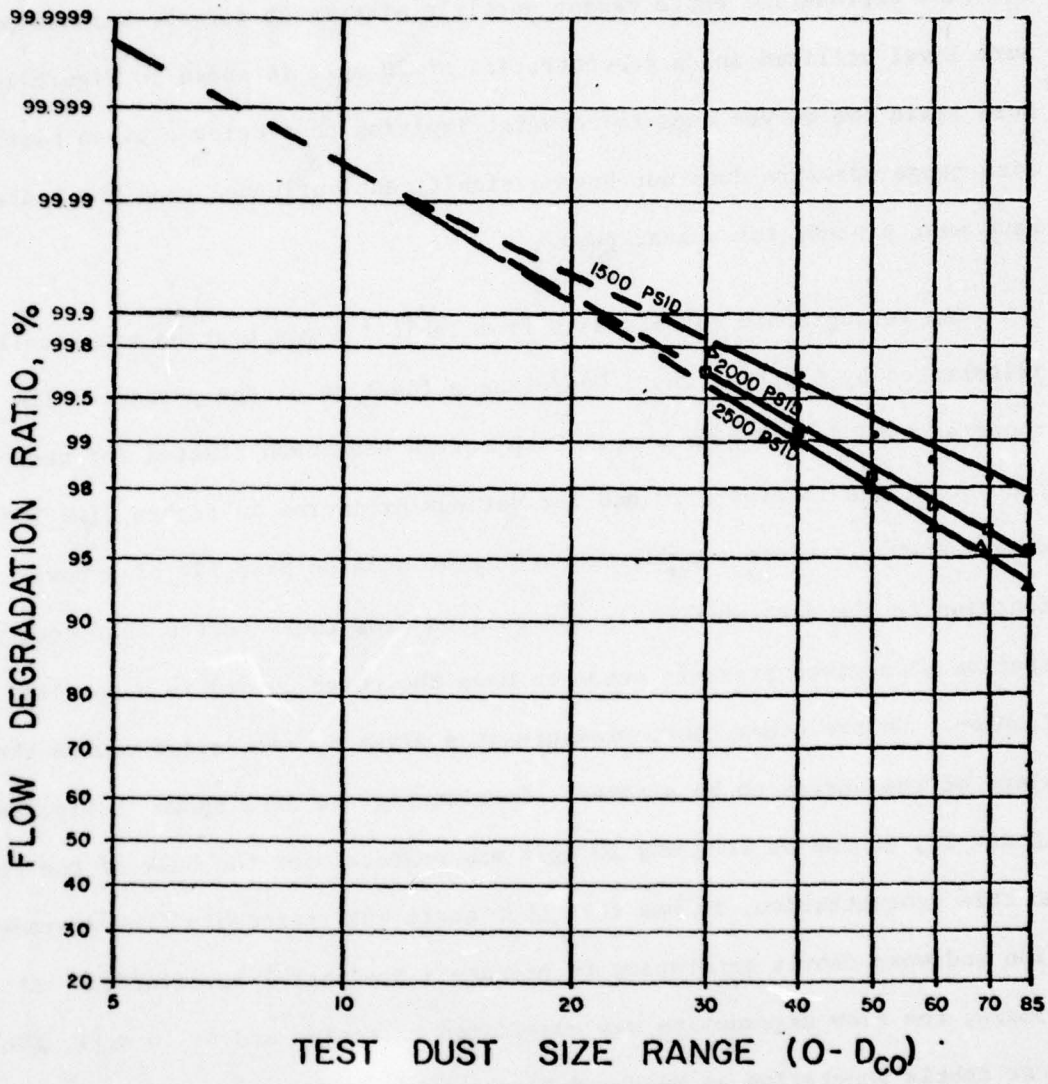


Figure 19. Flow Degradation Ratio as a Function of Particle Size Range for Various Pressures at 20 mg/l

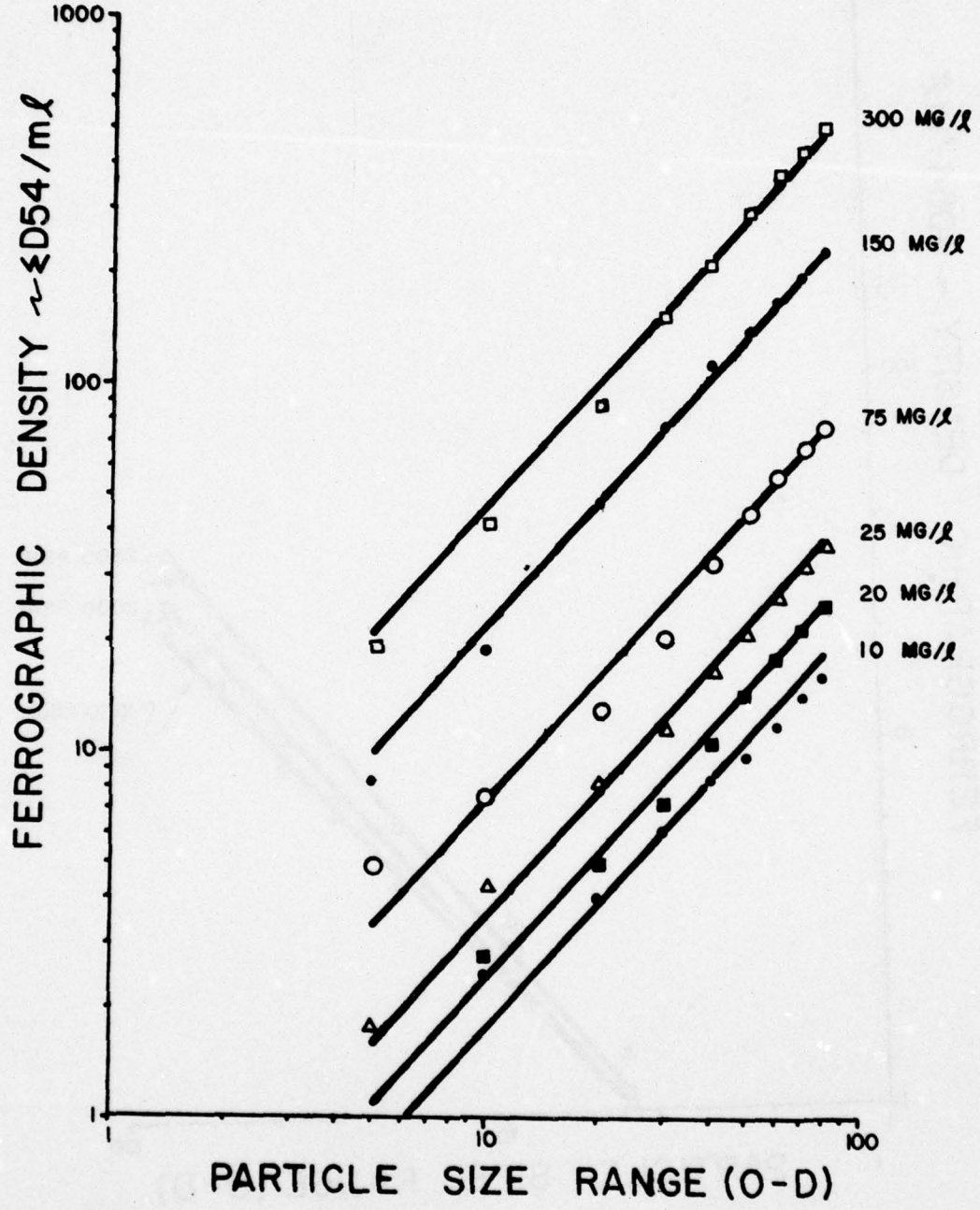


Figure 20. Ferrographic Density as a Function of Particle Size Range for Various Concentrations at 2000 psid

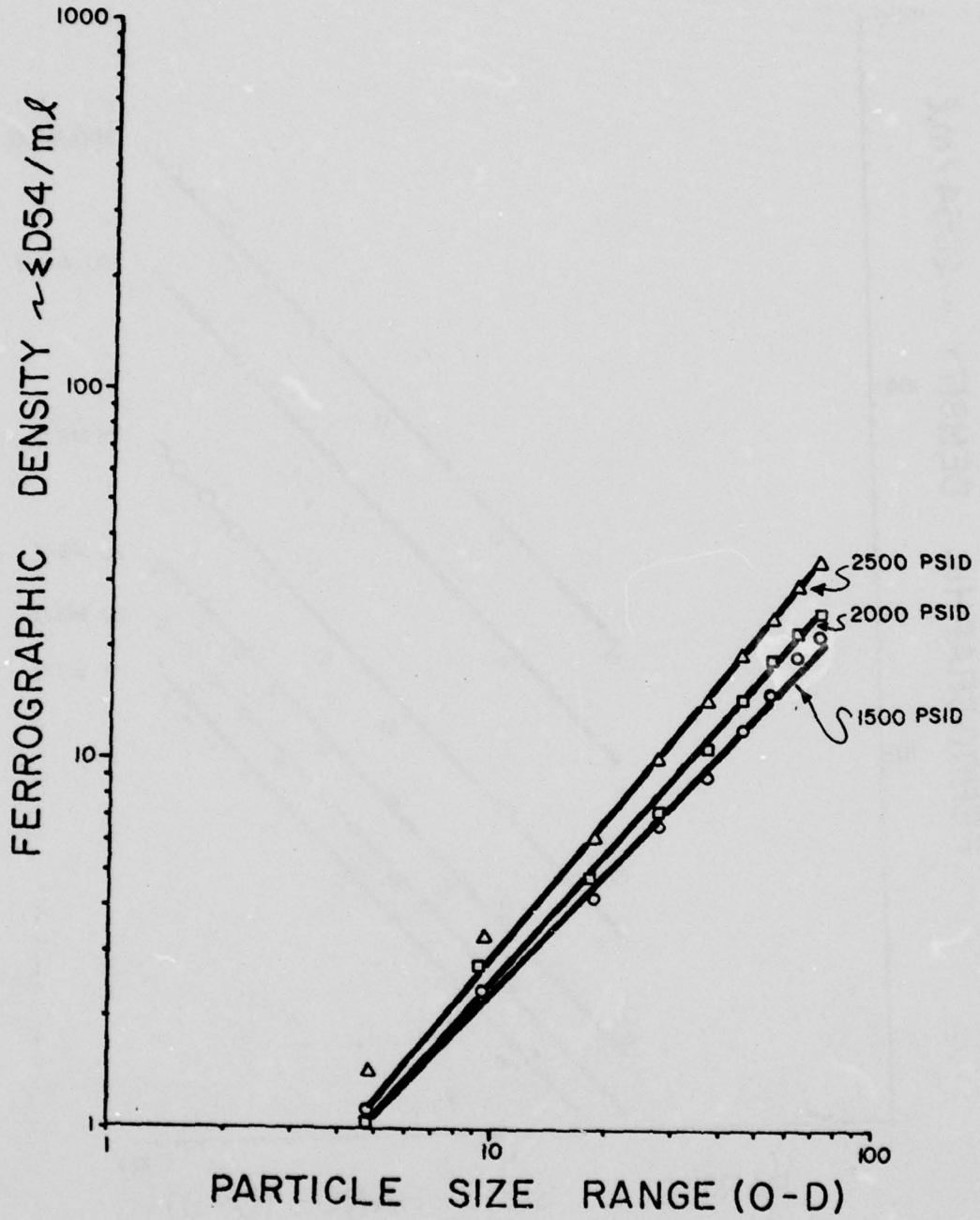


Figure 21. Ferrographic Density as a Function of Particle Size Range for Various Pressures at 20 mg/1

by actually fitting a power function to the test data as given in Table III. The equations developed for the various pressures and contaminant levels of the tests are shown in Table IV along with the coefficient of determination. As can be seen from Table IV the experimental data agreed closely with the theoretical equation form. These defined equations can be substituted into the equation of the Ferrographic Wear Model which describes the total flow change, ΔQ, as a function of the clearance change, Δh. In order to accomplish this it is necessary to use the expression which states that the accumulated Ferrographic density value is proportional to the equivalent clearance change, Δh, due to contaminant wear.

The equations given in Table IV were substituted into the Ferrographic Wear Model and the data shown in Table II was used to completely define the model. These results are shown graphically in Figures 22, 23, and 24 for tests conducted at 1500, 2000, and 2500 psid respectively. In these figures the solid lines were obtained from the model while the various symbols identified on each figure show the actual data values. The dashed part of each line shows the portion of the curve undefined by either the model or the data. Except for the data obtained at 10 mg/ℓ and 2000 psid (Figure 23) the model is a very good representation of the test data. The only explanation of the obvious trend reversal shown by the 10 milligram per litre curve in Figure 23 lies in the magnitude of both the Ferrographic data and the flow degradation results. There is considerable error introduced at such low levels and this was the primary reason for the use of 20 milligrams per litre in most of the tests.

It is felt that the comparison shown in Figures 22, 23, and 24 adequately demonstrates that model can be verified for a given gear pump. This

TABLE IV

SUMMARY OF EQUATION FOR FERROGRAPHIC DENSITIES

PRESSURE (PSID)	CONT. LEVEL MG/ℓ	EQUATION	COEF. OF DETER
1500	20	$y = 0.19\ D^{1.07}$	$r^2 = 0.991$
	150	$y = 2.20\ D^{1.07}$	$r^2 = 0.995$
2000	10	$y = 0.13\ D^{1.13}$	$r^2 = 0.997$
	20	$y = 0.17\ D^{1.13}$	$r^2 = 0.996$
	25	$y = 0.25\ D^{1.13}$	$r^2 = 0.998$
	75	$y = 0.52\ D^{1.13}$	$r^2 = 0.995$
	150	$y = 1.57\ D^{1.13}$	$r^2 = 0.997$
	300	$y = 3.29\ D^{1.13}$	$r^2 = 0.998$
2500	20	$y = 0.22\ D^{1.15}$	$r^2 = 0.996$
	150	$y = 2.12\ D^{1.15}$	$r^2 = 0.991$

$$y = \Sigma\frac{D54}{M\ell}$$

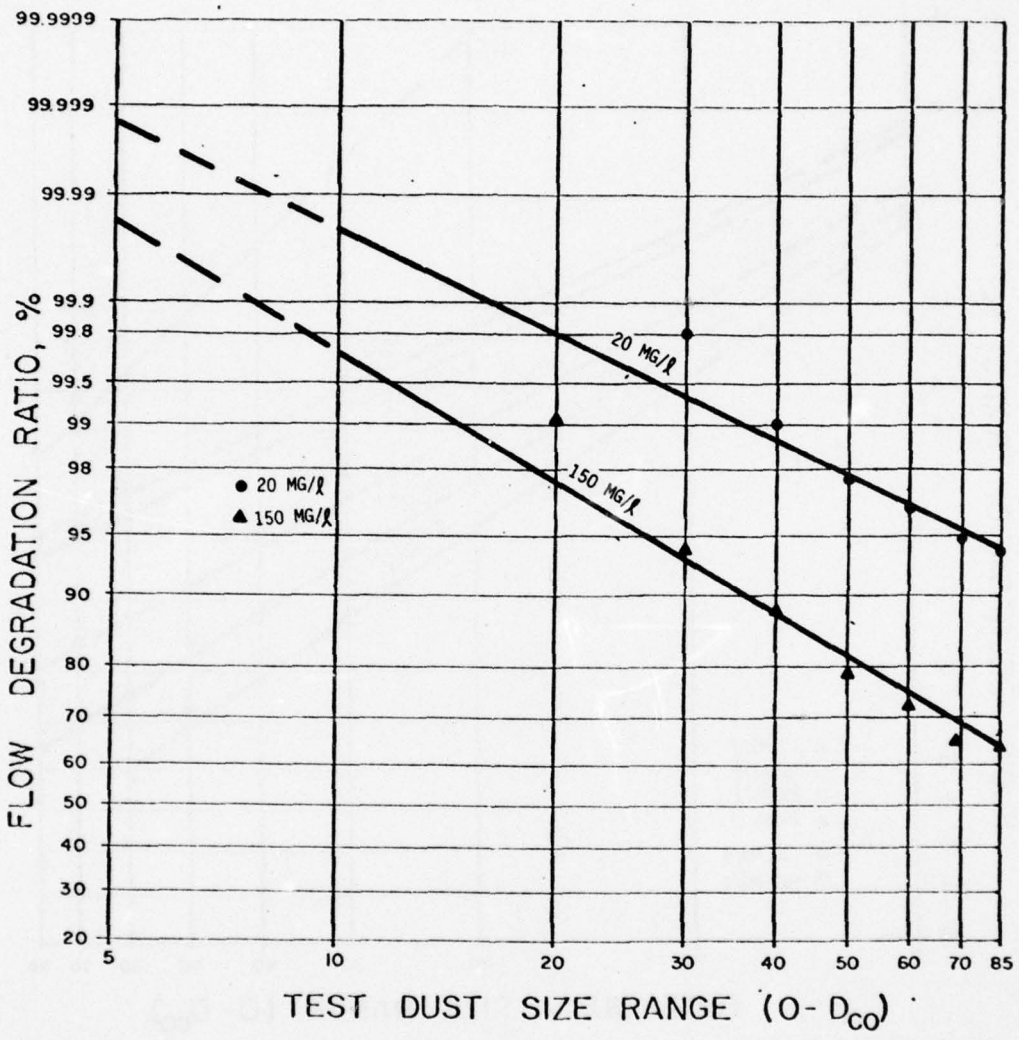


Figure 22. Comparison of Calculated and Measured Flow Degradation Ratio at 1500 psid

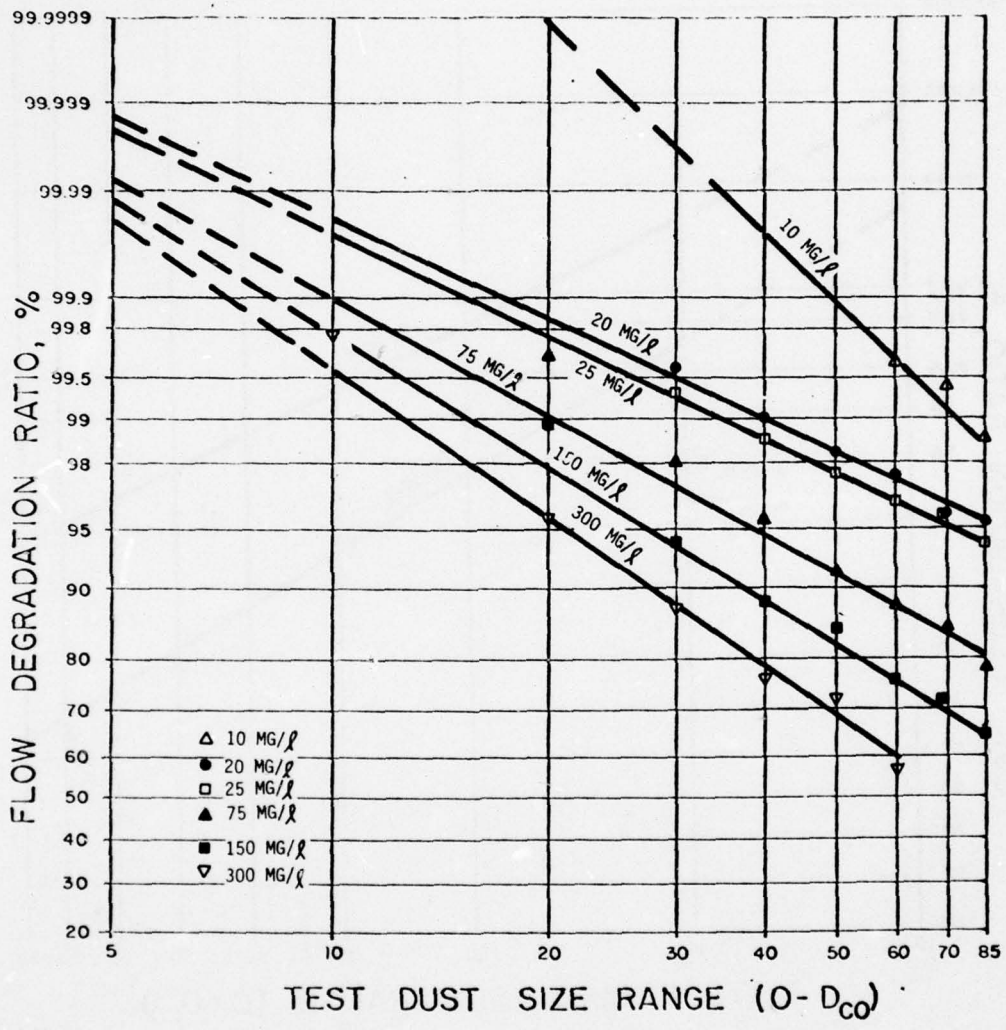


Figure 23. Comparison of Calculated and Measured Flow Degradation Ratio at 2000 psid

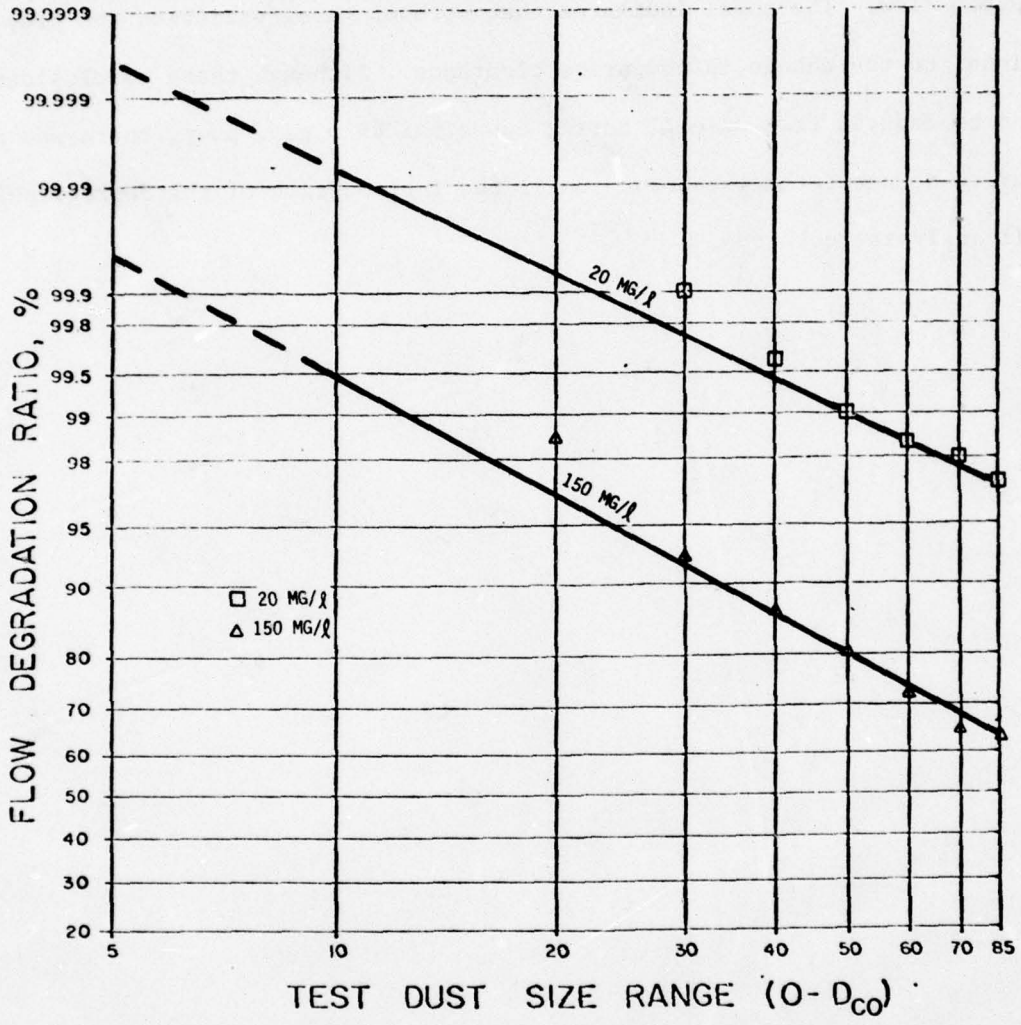


Figure 24. Comparison of Calculated and Measured Flow Degradation Ratio at 2500 psid

verification implies that the primary influence of contaminant wear is to increase the composite clearance of a gear pump. This change in clearance is reflected in two ways— generation of wear debris and degradation in output flow. The model indicates that both of these variables are proportional to the change in composite clearance. Although these conclusions can be deduced from an engineering appraisal of a gear pump, there was no way to document the phenomenon until the introduction of the Ferrographic oil analysis technique.

CHAPTER VI

SUMMARY AND CONCLUSION

Contaminant wear in hydraulic components has long been recognized as a major factor in the life and reliability of such components. Attempts to reduce the deliterious influence of particulate contamination entrained in the circulating fluid of a hydraulic system have taken two courses. Probably the most apparent solution to contaminant wear is the removal of these contaminant particles. Work in filtration of particulate contamination has produced much improvement in component life. The second approach to the problem of contaminant wear is concerned with the design of the components themselves. Through judicial design (configuration as well as materials) most hydraulic components can be made less sensitive to contaminant attack.

Improvements in the contaminant wear characteristics of hydraulic components have been hampered by two major obstacles. One stumbling block is associated with the inability to study the wear process directly while the second was the lack of a method to measure the results of the process in terms of the component function. Since, during operation of the component, the wear area is totally enclosed (to contain the fluid) any direct observations are impossible and disassembly is impractical. Therefore, measurement of the results of the wear process were forced to take an indirect form. The most widely accepted technique to evaluate contaminant wear is to measure the performance degradation due to the process. In order to accomplish this an accelerated basis the contamination level exposed to the component had to be high enough to produce sufficient

performance degradation to be readily and accurately measured. This requirement resulted in the use of 300 milligrams per litre in most contaminant sensitivity tests.

In hydraulic gear pumps such a high contamination level produced other problems. The most serious of these problems was the fact that, in most cases, the pump was totally destroyed during a test using only one set of operating conditions. If data was desired at any other condition (e.g., a higher or lower pressure) a new pump had to be used. This is both expensive and time consuming. With the introduction of the Ferrographic wear analysis technique, however, a new tool became available which offered a means of measuring the results of the contaminant wear process at much lower contamination levels. At these lower levels contaminant induced damage would be reduced and perhaps many tests could be conducted on the same pump.

Based upon the results of this research a modified contaminant sensitivity test can be developed for hydraulic pumps. The modification would include the reduction of the contamination level from 300 milligrams per litre to 20 milligrams per litre. Fluid samples would be extracted during each exposure for Ferrographic analysis. The proposed modified test would subject the various particle size ranges to the pump on a sequential basis. That is, a particle size range of 0-5 micrometres would be used first, followed by 0-10, 0-20, etc. micrometres. However, the test would continue in this manner only until three flow degradation points have been recorded.

Based upon experience, it would be expected that the modified test would be conducted until a particle size range of 0-40 or 0-50 had been exposed to the pump. At this point in the test, the pump would have exhibited a

measurable flow degradation at three contaminant exposures. Furthermore, Ferrographic density data would be available at 5 or 6 exposures. This would provide sufficient data to define the Ferrographic Wear Model which then could be used to predict the degradation at all nine contaminant particle size ranges. It should be noted that the total flow degradation exhibited by the pumps tested as part of the effort after exposure to 0-50 micrometre size range was approximately 2%. The operating pressure of the pump could then be increased or decreased to obtain similar data at a different pressure. This whole process could be repeated, at least on the pumps tested here, at four different pressures and the total flow degradation would not exceed 8-10%.

The fact that the test data presented here failed to show a significant influence from pump speed is important. This implies that speed is not a parameter of concern in the contaminant wear characteristics of a hydraulic gear pump. The true advantage of this discovery lies in the fact that very large pumps can be tested at lower than rated speed without unduly influencing the contaminant wear evaluation. In addition, the number of tests required on a given pump can be reduced to only pressure consideration. Therefore, a pump which produces 100 gallons per minute at 2500 revolutions per minute could be evaluated on a test facilities with a maximum flow capability of 60 gallons per minute by merely conducting the test at a speed lower than 2500 RPM.

Another important factor resulting from the work accomplished by this effort is that the test sequence does not significantly influence the outcome. Therefore, if it is desired to evaluate a pump at four operating

pressures, the evaluation at one pressure does not bias the results at another pressure. This assumes, of course, that the total flow degradation of the pump is not excessive. In addition, if a pump is assessed at, say four pressures, and some time later a new pump is evaluated at a different set of pressure levels the results of both tests should correlate provided the design of the pump has not been altered.

It is felt that through the work presented here, the Ferrographic wear analysis technique has been shown to be an effective method for evaluating contaminant wear. The isolation mechanisms used in this process are ideally suited to contaminant wear tests. In such experiments, test contaminants are intentionally injected into the circulating fluid to produce a control contaminant wear process. The analysis or measurement of the wear debris generated during the test would be impossible unless the debris can be separated from the introduced contamination. The correlation obtained between Ferrographic density and flow degradation data through consideration of the clearance change indicates that the Ferrographic technique is an effective means of isolating the wear debris from the extraneous material. Once separated a great deal of information can be obtained from the wear debris. Although not included here, the size and shape of the wear debris can be used to deduce facts concerning the nature of the contaminant attack. Furthermore, elemental analysis of the debris collected on a Ferrogram can be performed to learn the major constituent present. However, such appraisals are more rewarding when the wear process is not controlled.

The fundamental objective of this effort was based upon the fact that the primary influence of contaminant wear in a gear pump is the enlargement

of the composite clearances within the pump. This increase in clearance is reflected by both the generation of wear debris and the loss of output flow at a given pressure. At the initiation of this effort, it was proposed that the amount of wear debris generated was directly proportional to the change in pump clearance. Thus by measuring the wear debris concentration in the circulating fluid after exposing the pump to a given and controlled contaminant environment, the resulting increase in clearance could be assessed.

The Ferrographic wear analysis method was used to appraise the wear debris concentration. However, since it was impossible to measure either the initial clearances or the change of clearance within a pump, a second parameter (flow degradation) was used to evaluate the hypothesis. This required the development of a Ferrographic Wear Model which included both the Ferrographic parameters as well as flow degradation. Thus, the Ferrographic density data was used to develop a flow degradation model which in turn was fit to the flow degradation data. The goodness of fit between the model and the data was used as a criterion for acceptability of the model.

There were 24 pump contaminant wear tests conducted in this effort along with five pump performance tests and numerous experiments associated with the Ferrographic technique. The results of the performance tests were used to determine the form of the flow degradation model. The Ferrographic experiments were conducted to evaluate the capabilities of the technique in terms of linearity, saturation and repeatability. Initial tests with the Ferrograph reveal a need for improvement which was accomplished. During the pump contaminant wear tests, samples were extracted at each contaminant

exposure and the output flow was carefully measured. The pumps were evaluated using the following parameters as test variables.

1. Outlet pressure - 1500, 2000, 2500 psi
2. Shaft speed - 1500, 2000, 2500 RPM
3. Contaminant concentration (10, 20, 25, 75, 250, 300 milligrams per litre)
4. Particle size range (0-5, 0-10, 0-20, 0-30, 0-40, 0-50, 0-60, 0-70, 0-80 micrometres)

The test plan was formulated in such a manner that the influence of pump speed, outlet pressure, and testing sequence could all be assessed through standard statistical techniques. It was proposed that by using wear debris analysis the contamination level of the standard wear test could be drastically reduced. At this reduced contaminant concentration destruction due to contaminant wear would also be reduced and more than one test could be conducted on the same pump specimen. Thus, the test sequence aspect refers to the influence of conducting more than one contaminant wear test on the same pump. The results of the statistical analysis revealed that speed and sequence had insignificant effect on the test results but that pressure could not be ignored. Thus, the results of tests conducted in various sequences and at different speeds were averaged with the other test data to provide a stronger data base.

The Ferrographic density data (ΣD54/ml) was used to formulate expressions which described the wear debris generation as a function of the significant variables of the testing program. This phase of effort revealed that the amount of debris generated versus particle size range was an

excellent fit to an exponential model (log-log). The coefficient of this particular power function was found to be related to the concentration in milligrams per litre of the contaminant environment while the exponent indicated a dependency upon the outlet pressure of the test pump. The Ferrographic density values were assumed to be proportional to the clearance change within the pump and the flow degradation data was utilized in determining the constant or proportionally. Therefore, a complete Ferrographic Wear Model was developed and compared to the test data. This comparison revealed excellent correlation between the flow degradation ratios calculated from the model and the ratios derived directly from the actual output flow data.

Conclusions

The accomplishments of this effort have contributed significantly to the area of contaminant wear. Prior to this work the use of wear debris to evaluate the contaminant wear of a hydraulic component had not been successfully achieved. Through the use of the Ferrographic oil analysis technique, this effort has shown that the amount of debris generated by a contaminant wear process can be used to evaluate the degree of wear in a gear pump. In addition, since flow loss is the normal manner used by the engineering world for expressing wear in a hydraulic gear pump, this research work has shown that the amount of wear debris generated will correlate with the accompanying flow degradation through proper consideration of the clearance changes.

The noteworthy contributions resulting from this effort can be outlined as follows:

1. As a result of the extensive modifications to the Ferrograph operating technique and the associated equipment made as a necessary part of this study, a repeatable and viable wear debris analysis method was evolved.
2. A major discovery resulting from this work is that the amount of wear debris generated from a gear pump due to contaminant exposure is a power function of the particle size range exposed and that:
   a. The coefficient of this power function is dependent upon the concentration of the contaminant environment which produced the debris, and that
   b. The exponent of this power function is related to outlet pressure imposed.
3. Based upon the results of actual pump tests conducted in accordance with a factorially arranged test plan, pump speed was shown to have no significant influence upon the contaminant wear of a gear pump.
4. The Ferrographic Wear Model developed and advanced in this study has been shown to accurately characterize the relationships for wear debris generation and flow degradation associated with a gear pump.
5. It was demonstrated by this effort that by using a fluid environment having a contaminant concentration of only 20 milligrams/litre, a single pump can be sequentially (at least three times) tested at different operating conditions without significantly biasing the test results. This means that the contaminant

tolerance of a gear pump can be completely defined over a broad spectrum of operating conditions using a single pump.

6. A practical and significant engineering contribution to international fluid power standardization was made by this research effort in providing a firm basis for a new contaminant sensitivity test in which a drastically reduced contamination level (20 MG/ℓ as compared to 300 MG/ℓ in the standard test) is used with a single pump to assess its tolerance at different operating conditions (pressure, temperature, etc.).

CHAPTER VII

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APPENDIX A

PUMP PERFORMANCE DATA

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The output flow at various outlet pressures and shaft speeds were measured for five pumps. The pumps utilized for this phase of effort carried the following identification number 300, 301, 302, 303, 311. Pump No. 300 was evaluated at 2000 psi outlet pressures while pump No. 301 was tested at 1500, 2000, and 2500 psi. Pumps 302 and 303 were appraised at 500, 1000, 1500, 2000, and 2500 psi. In all of these tests, output flow was measured at both increasing and decreasing speeds with an average result used. Pump No. 311 was tested at constant speeds of 1500, 2000, and 2500 RPM while the pressure was varied from 600 to 2500 psi. The following tables show the actual flow results obtained.

TABLE A1

OUTPUT FLOW DATA FOR PUMP NO. 300

SPEED (RPM)	OUTPUT FLOW AT PRESSURE
	2000 psi
1000	5.50
1100	6.19
1200	6.84
1300	7.50
1400	8.15
1500	8.81
1600	9.47
1700	10.14
1800	10.79
1900	11.47
2000	12.09
2100	12.77
2200	13.44
2300	14.09
2400	14.74
2500	15.39

TABLE AII

OUTPUT FLOW DATA FOR PUMP NO. 301

SPEED (RPM)	OUTPUT FLOW AT INDICATED PRESSURE (GPM)		
	1500 psi	2000 psi	2500 psi
1000	5.76	5.63	5.49
1100	6.41	6.29	6.06
1200	7.09	6.96	6.73
1300	7.71	7.62	7.39
1400	8.41	8.26	8.05
1500	9.05	8.94	8.73
1600	9.67	9.54	9.37
1700	10.34	10.18	10.07
1800	11.00	10.85	10.70
1900	11.65	11.50	11.37
2000	12.30	12.17	11.99
2100	12.96	12.82	12.71
2200	13.59	13.48	13.33
2300	14.22	14.15	13.98
2400	14.88	14.80	14.64
2500	15.52	15.42	15.26

TABLE AIII

OUTPUT FLOW DATA FOR PUMP NO. 302

SPEED (RPM)	OUTPUT FLOW AT INDICATED PRESSURE (GPM)				
	500 psi	1000 psi	1500 psi	2000 psi	2500 psi
1000	6.19	5.97	5.86	5.63	5.48
1100			6.49	6.30	6.15
1200			7.13	6.94	6.75
1300			7.77	7.62	7.41
1400			8.43	8.24	8.05
1500	9.39	9.22	9.03	8.88	8.69
1600			9.71	9.56	9.33
1700			10.34	10.22	10.03
1800			10.98	10.87	10.68
1900			11.65	11.52	11.33
2000	12.57	12.42	12.29	12.19	12.00
2100			12.94	12.82	12.65
2200			13.61	13.44	13.34
2300			14.26	14.11	13.99
2400			14.94	14.72	14.65
2500	15.74	15.67	15.57	15.44	15.30

TABLE AIV

OUTPUT FLOW DATA FOR PUMP NO. 303

SPEED (RPM)	OUTPUT FLOW AT INDICATED PRESSURE (GPM)				
	500 psi	1000 psi	1500 psi	2000 psi	2500 psi
1000	6.03	5.73	5.46	5.13	4.59
1100			6.10	5.78	5.19
1200			6.75	6.40	5.97
1300			7.43	7.07	6.64
1400			8.01	7.75	7.30
1500	9.18	8.95	8.63	8.39	7.96
1600			9.30	9.11	8.67
1700			10.10	9.65	9.31
1800			10.66	10.30	9.96
1900			11.33	10.96	10.62
2000	12.36	12.15	11.94	11.56	11.31
2100			12.59	12.23	11.96
2200			13.23	12.92	12.65
2300			13.86	13.65	13.30
2400			14.49	14.30	13.92
2500	15.59	15.36	15.13	15.01	14.55

TABLE AV

OUTPUT FLOW DATA FOR PUMP NO. 311

OUTLET PRESSURE (PSI)	OUTPUT FLOW AT INDICATED SPEED (GPM)			
	1500 RPM	2000 RPM	2500 RPM	
2500	9.31	12.58	16.15	
2400	9.35	12.59	16.18	
2300	9.37	12.61	16.17	
2200	9.41	12.61	16.17	
2100	9.43	12.61	16.18	
2000	9.45	12.65	16.18	
1900	9.48	12.67	16.22	
1800	9.52	12.69	16.24	
1700	9.54	12.73	16.28	
1600	9.54	12.73	16.30	
1500	9.58	12.77	16.32	
1400	9.60	12.77	16.34	
1300	9.62	12.80	16.36	
1200	9.64	12.84	16.36	
1100	9.66	12.84	16.39	
1000	9.69	12.88	16.41	
900	9.73	12.90	16.41	
800	9.73	12.94	16.43	
700	9.77	12.94	16.45	
600	9.81	12.97	16.51	

APPENDIX B

PUMP CONTAMINANT WEAR TEST AND DESCRIPTION OF CLASSIFIED TEST DUST

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The following procedure was followed in conducting the contaminant wear tests for the research investigation. Essentially, this procedure is the same as the pump contaminant sensitivity test [49]. The only major exceptions are that the contamination level used was not always the 300 milligrams per litre called out in the standard and all pumps were subjected to a break in procedure at 2500 RPM and 2500 psi. The standard specifies that the break in be conducted at the conditions of the test. Many of the steps given in the following procedure were excerpted directly from the standard test [49].

1.0 PREPARATION OF TEST EQUIPMENT

1.1 Install a pump which is known to be relatively insensitive to contamination in the test circuit.

1.2 Adjust the system volume so that it equals one-fourth (±10 percent) of the lowest volume flow rate to be used for testing.

1.3 Circulate fluid thru the filter until the contaminant background is less than 10 mg/ℓ.

1.4 By-pass the filter.

1.5 Add AC Fine Test Dust to the system to bring the contamination level to 300 mg/ℓ (± 10 mg/ℓ).

1.6 Inject the contaminant in the form of a well-mixed slurry to prevent agglomeration of particles, ensuring that injection remains uniform over a one minute period.

1.7 Operate the system at the minimum flow rate to be used for testing.

1.8 Extract four fluid samples at 15 minute intervals from the system.

1.9 Measure the gravimetric level of each sample.

1.10 Circulate fluid thru the filter until the contaminant background is is less than 10 mg/ℓ.

1.11 Consider the system qualified for testing if the gravimetric levels of clause 1.9, as obtained in clause 1.8's samples, are within ± 10% of the initial gravimetric level requirement of clause 1.5.

2.0 TEST PROCEDURE

2.1 Install the test pump in the circuit.

2.2 Adjust the system volume so that it equals one-fourth (± 10 percent) of the volume flow rate of the pump to be tested, exclusive of the filter cleanup circuit.

2.3 Operate the pump at the 150° F and 2500 RPM with the filter in the circuit. Use the following schedule for the pump operating pressure:

15 minutes at 25% of 2500 psi

15 minutes at 50% of 2500 psi

15 minutes at 75% of 2500 psi

2.4 Operate at 2500 psi until flow rate has remained constant for at least 60 minutes.

2.5 Record the flow rate at the specified pressure and speed as the rated flow of the pump (Q_r).

2.6 Determine the quantity of contaminant (g_i) required for each injection.

2.7 Prepare a slurry containing g_i grams of contaminant in each of the following size ranges: 0-5, 0-10, 0-20, 0-30, 0-40, 0-50, 0-60, 0-70, and 0-80 micrometres.

2.8 Operate the pump at specified test conditions.

2.9 By-pass the filter.

2.10 Inject the 0-5 μm slurry uniformly over a period of one minute.

2.11 Observe and record the flow rate at two-minute intervals.

2.12 Continue operating until the flow rate becomes constant for 10 minutes or until 30 minutes have elapsed, whichever occurs first.

2.13 Circulate fluid thru the filter for 10 minutes.

2.14 Record the final flow rate for the injection with the pump operating at specified test conditions. If the flow rate has decreased to less than 70% of its original rated value (Q_r), the test can be terminated.

2.15 Repeat clauses 2.9 up to and including 2.14 using 0-10, 0-20, 0-30, 0-40, 0-50, 0-60, 0-70, and 0-80 μm contaminant in progressively increasing sizes.

DESCRIPTION OF CLASSIFIED TEST DUST

The classified test dust specified in this recommended standard is to be obtained by a recognized commercial classifier using AC Fine Test Dust as the base stock. Operate the classifier in accordance with the manufacturer's recommendations. The amount (by weight) of dust collected in the classified fraction at the end of the classification procedure is to be within the following limits:

Classified Size (μm)	Weight Percentage of Classified Fraction Relative to Full Distribution of AC Fine Test Dust		
	Minimum %	Average %	Maximum %
0-5	36.0	39.0	42.0
0-10	54.0	57.0	60.0
0-20	70.0	73.0	76.0
0-30	82.2	85.2	88.2
0-40	88.0	91.0	94.0
0-50	91.0	94.0	97.0
0-60	93.0	96.0	99.0
0-70	94.2	97.2	100.0
0-80	95.0	98.0	100.0

APPENDIX C

FLOW DEGRADATION DATA

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TABLE CI

OUTPUT FLOW RESULTS FROM CONTAMINANT WEAR TESTS

Speed (RPM)	Pressure (psi)	Cont.Level mg/l	Q_i (GPM)	Output Flow after Indicated Exposure (GPM)								
				0-5	0-10	0-20	0-30	0-40	0-50	0-60	0-70	0-80
1500	1500	20	7.99	7.99	7.99	7.99	7.99	7.96	7.92	7.84	7.80	7.61
	2000	20	8.30	8.30	8.30	8.30	8.30	8.26	8.22	8.15	8.11	7.99
	2500	20	7.65	7.65	7.65	7.65	7.65	7.54	7.46	7.35	7.12	7.04
2000	1500	20	10.57	10.57	10.57	10.57	10.41	10.34	10.30	10.22	10.15	10.03
		20	11.44	11.44	11.44	11.44	11.44	11.41	11.37	11.33	11.15	11.04
		20	11.29	11.29	11.29	11.29	11.29	11.25	11.21	11.18	11.14	11.10
		150	11.40	11.40	11.40	11.25	10.68	9.88	9.16	8.21	7.53	7.34
	2000	0	11.25	11.25	11.25	11.25	11.25	11.25	11.25	11.25	11.25	11.25
		10	11.25	11.25	11.25	11.25	11.25	11.25	11.25	11.21	11.18	11.10
		20	11.10	11.10	11.10	11.10	11.10	11.10	10.95	10.87	10.80	10.64
		20	11.06	11.06	11.06	11.06	10.99	10.91	10.83	10.80	10.70	10.64
		20	10.76	10.76	10.76	10.76	10.72	10.68	10.64	10.57	10.53	10.45
		25	12.32	12.32	12.32	12.32	12.28	12.16	12.05	11.90	11.78	11.59
		75	11.82	11.82	11.82	11.78	11.59	11.25	10.87	10.45	10.03	9.43
		150	12.28	12.28	12.28	12.16	11.59	10.87	10.34	9.39	8.74	8.02
		300	12.51	12.51	12.47	12.09	10.95	10.15	9.35	7.7		
		300	12.39	12.39	12.35	11.78	10.83	9.01	8.25	6.8		
	2500	20	10.72	10.72	10.72	10.72	10.72	10.57	10.45	10.26	10.15	9.54
		20	11.48	11.48	11.48	11.48	11.48	11.48	11.40	11.29	11.10	10.95
		20	11.56	11.56	11.56	11.56	11.52	11.37	11.21	11.10	10.95	10.83
		150	11.80	11.80	11.80	11.71	11.18	10.34	9.39	8.46	7.73	7.50
2500	1500	20	15.12	15.12	15.12	15.12	15.09	15.05	14.97	14.93	14.86	14.78
	2000	20	14.32	14.32	14.32	14.32	14.21	14.13	13.98	13.87	13.64	13.49
	2500	20	14.70	14.70	14.70	14.70	14.67	14.59	14.40	14.32	14.10	13.94

TABLE CII

FLOW DEGRADATION RESULTS FROM CONTAMINANT WEAR TESTS

Speed (RPM)	Pressure	Cont. Level mg/l	Q_i (GPM)	Flow Degradation after indicated exposure (GPM) $Q_f - Q_i$								
				0-5	0-10	0-20	0-30	0-40	0-50	0-60	0-70	0-80
1500	1500	20	7.99	0	0	0	0	0.03	0.07	0.15	0.19	0.38
	2000	20	8.30	0	0	0	0	0.04	0.08	0.15	0.19	0.31
	2500	20	7.65	0	0	0	0	0.11	0.19	0.30	0.53	0.81
2000	1500	20	10.57	0	0	0	0.16	0.23	0.27	0.35	0.42	0.54
		20	11.44	0	0	0	0	0.03	0.07	0.11	0.29	0.40
		20	11.29	0	0	0	0	0.04	0.08	0.11	0.15	0.19
		150	11.40	0	0	0.15	0.72	1.52	2.24	3.19	3.87	4.06
	2000	0	11.25	0	0	0	0	0	0.04	0	0	0
		10	11.25	0	0	0	0	0	0	0.04	0.07	0.15
		20	11.10	0	0	0	0	0	0.15	0.23	0.30	0.46
		20	11.06	0	0	0	0.07	0.15	0.23	0.26	0.36	0.42
		20	10.76	0	0	0	0.04	0.08	0.12	0.19	0.23	0.31
		25	12.32	0	0	0	0.04	0.16	0.27	0.42	0.54	0.73
		75	11.82	0	0	0.04	0.23	0.57	0.95	1.37	1.79	2.39
		150	12.28	0	0	0.12	0.69	1.41	1.94	2.89	3.54	4.26
		300	12.51	0	0.04	0.42	1.56	2.36	3.16	4.81		
		300	12.39	0	0.04	0.61	1.56	3.38	4.14	5.59		
	2500	20	10.72	0	0	0	0	0.15	0.27	0.46	0.57	1.18
		20	11.48	0	0	0	0	0	0.08	0.19	0.38	0.53
		20	11.56	0	0	0	0.04	0.19	0.35	0.46	0.61	0.73
		150	11.80	0	0	0.09	0.62	1.42	2.41	3.34	4.07	4.30
2500	1500	20	15.12	0	0	0	0.03	0.07	0.15	0.19	0.26	0.34
	2000	20	14.32	0	0	0	0.11	0.19	0.34	0.45	0.68	0.83
	2500	20	14.70	0	0	0	0.00	0.11	0.30	0.38	0.60	0.76

TABLE CIII

FLOW DEGRADATION RATIO RESULTS FROM CONTAMINANT WEAR TESTS

Speed (RPM)	Pressure (psi)	Cont. Level mg/l	Q_i (GPM)	Flow Degradation Ratio after Indicated Exposure (Q_f/Q_i) 0-5	0-10	0-20	0-30	0-40	0-50	0-60	0-70	0-80
	1500	20	7.99	1.0	1.0	1.0	1.0	.996	.991	.981	.976	.952
	2000	20	8.30	1.0	1.0	1.0	1.0	.995	.991	.982	.977	.963
	2500	20	7.65	1.0	1.0	1.0	1.0	.994	.983	.960	.930	.920
	1500	20	10.57	1.0	1.0	1.0	.985	.978	.974	.967	.960	.950
		20	11.44	1.0	1.0	1.0	1.0	.997	.994	.990	.975	.965
		20	11.29	1.0	1.0	1.0	1.0	.996	.993	.990	.987	.983
		150	11.40	1.0	1.0	.987	.937	.867	.803	.720	.660	.643
	2000	0	11.25	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
		10	11.25	1.0	1.0	1.0	1.0	1.0	1.0	.996	.994	.987
		20	11.10	1.0	1.0	1.0	1.0	1.0	.986	.979	.973	.959
		20	11.06	1.0	1.0	1.0	.994	.986	.979	.976	.967	.962
		20	10.76	1.0	1.0	1.0	.996	.993	.989	.982	.979	.971
2000		25	12.32	1.0	1.0	1.0	.997	.987	.978	.966	.956	.941
		75	11.82	1.0	1.0	.997	.981	.952	.920	.884	.849	.798
		150	12.28	1.0	1.0	.990	.944	.885	.842	.765	.712	.653
		300	12.51	1.0	.997	.966	.875	.811	.747	.616		
		300	12.39	1.0	.997	.952	.874	.727	.666	.549		
	2500	20	10.72	1.0	1.0	1.0	1.0	.986	.975	.957	.947	.890
		20	11.48	1.0	1.0	1.0	1.0	1.0	.993	.983	.967	.954
		20	11.56	1.0	1.0	1.0	.997	.984	.970	.960	.947	.937
		150	11.80	1.0	1.0	.992	.947	.876	.796	.717	.655	.636
	1500	20	15.12	1.0	1.0	1.0	.998	.995	.990	.987	.983	.978
2500	2000	20	14.32	1.0	1.0	1.0	.992	.987	.976	.969	.953	.942
	2500	20	14.70	1.0	1.0	1.0	.998	.993	.980	.974	.959	.948

APPENDIX D

FERROGRAPHIC DENSITY DATA

TABLE DI

SUMMARY OF FERROGRAPHIC DENSITY DATA

Speed (RPM)	Pressure (psi)	Conc.Level mg/l	Q_i (GPM)	Ferrographic Density Data after Indicated Exposure (D54/ml)							
				0-5	0-10	0-20	0-30	0-40	0-50	0-60	0-70	0-80
1500	1500	20	7.99	0.83	1.72	3.15	5.05	7.17	9.22	12.95	15.67	18.34
	2000	20	8.30	1.05	3.03	5.83	8.31	12.88	16.71	20.64	24.37	28.64
	2500	20	7.65	0.98	2.96	5.71	9.43	13.65	17.39	22.09	27.19	31.06
2000	1500	20	10.57	1.09	2.38	3.94	5.81	8.03	10.84	13.78	17.09	20.78
		20	11.44	1.40	2.64	5.07	8.17	10.95	13.58	15.80	18.80	22.43
		20	11.29	1.42	3.04	5.66	7.83	11.23	13.86	16.76	19.83	22.46
		150	11.40	7.95	16.85	44.85	88.52	120.85	155.18	206.85	254.52	294.85
	2000	0	11.25	0.50	1.15	1.63	2.18	2.74	3.23	3.61	3.89	4.22
		10	11.25	1.11	2.52	4.33	6.04	7.53	9.24	11.44	13.27	15.22
		20	11.10	1.30	3.23	4.83	7.14	9.76	11.99	16.61	19.81	24.16
		20	11.06	0.91	2.24	4.61	7.86	11.39	16.06	19.83	24.46	27.89
		20	10.76	1.34	2.87	4.84	7.41	10.41	13.71	16.28	19.91	24.54
		25	12.32	1.75	4.28	8.21	11.46	16.13	19.58	24.35	28.20	34.40
		75	11.82	3.85	7.52	12.95	21.32	33.32	44.92	56.52	68.52	78.52
		150	12.28	8.4	19.0	47.67	77.17	118.34	141.17	174.47	201.30	236.63
		300	12.51	25.0	48.50	81.0	139.83	202.50	318.33	392.5		
		300	12.39	13.4	37.0	93.0	165.0	234.33	291.0	359.67		
	2500	20	10.72	1.42	3.12	4.43	6.74	11.12	17.89	24.93	31.10	37.57
		20	11.48	1.42	3.70	8.38	12.53	17.10	22.13	27.70	34.43	39.70
		20	11.56	1.94	3.47	5.95	9.15	13.15	16.65	19.22	23.19	28.32
		150	11.80	5.0	19.8	51.8	86.8	151.47	202.8	272.47	333.14	415.14
2500	1500	20	15.12	1.20	2.73	5.01	7.33	10.01	14.08	18.61	21.54	25.11
	2000	20	14.32	0.93	2.55	3.97	6.02	9.55	14.15	17.15	22.42	25.99
	2500	20	14.70	1.68	3.91	6.81	11.21	16.74	21.41	28.11	36.52	-

APPENDIX E

STATISTICAL ANALYSIS CALCULATIONS

The statistical arrangements of the tests in the research investigation permit evaluation of three parameters. Since more than one test was conducted on a given pump, a latin square arrangement was included to appraise the influence of repeated testing on a single pump. In addition, a factorially arranged segment of the test plan allows assessment of the statistical significance of both pressure and speed. Since both flow degradation and Ferrographic data was collected, statistical analysis can be performed on both sets of data. Furthermore, consideration of the data will reveal that the most effect of any variable included in the test plan will be at the 0-80 micrometre particle size range exposure. That is if a variable indicates a non-significant influence at 0-80, it will be even less significant at small size ranges.

TEST SEQUENCE

FLOW DEGRADATION RATIO AT 0-80

Pressure (PSI)	Test No. 1	Test No. 2	Test No. 3	
1500	0.965	0.983	0.950	2.898
2000	0.959	0.962	0.971	2.892
2500	0.939	0.890	0.954	2.783
	2.863	2.835	2.875	8.573

PUMPS $\frac{300}{2.799}$ $\frac{306}{2.881}$ $\frac{308}{2.893}$

$$C.F. = \frac{(8.573)^2}{3^2} = 8.166$$

$$\text{TOTAL} = (.965)^2 = (.983)^2 + \ldots + (.890)^2 + (.954)^2 - C.F. = .006$$

$$\text{ROW (PRESS)} = \frac{1}{3}[2.898^2 + 2.892^2 + 2.783^2] - C.F. = .003$$

$$\text{COLUMN (SEQ.)} = \frac{1}{3}[2.863^2 + 2.835^2 + 2.875^2] - C.F. = .00054$$

$$\text{PUMP} = \frac{1}{3}[2.799^2 + 2.881^2 + 2.893^2] - C.F. = .002$$

ANALYSIS OF VARIANCE

SOURCE	D.F.	S.S	M.S	F_c	$F_{t(5\%)}$
Row(Press	2	.003	.0015	6.52	
Column(Seq)	2	.00054	.00027	1.17	19.0
Pump	2	.002	.0010	4.35	
Error	2	.00046	.00023		

Sequence is not significant at the 5% level

TEST SEQUENCE

FERROGRAPHIC DENSITY at 0-80

Pressure (PSI)	Test No. 1	Test No. 2	Test No. 3	
1500	22.43	22.46	20.78	65.67
2000	24.16	27.89	25.54	76.59
2500	28.32	37.57	39.70	107.59
	74.91	87.92	85.02	247.85

PUMPS $\frac{300}{82.51}$ $\frac{306}{90.02}$ $\frac{308}{75.82}$

$$C.F. = \frac{(247.85)^2}{3^2} = 6825.514$$

$$TOTAL = (22.43)^2 + (22.46)^2 + \ldots (37.57)^2 + (39.7)^2 = 367.238$$

$$ROW\ (PRESS) = \frac{1}{3}[65.67^2 + 76.59^2 + 105.59^2] - C.F. = 283.761$$

$$COLUMN\ (SEQ.) = \frac{1}{3}[74.91^2 + 87.92^2 + 85.02^2] - C.F. = 31.097$$

$$PUMPS = \frac{1}{3}[82.5^2 + 90.02^2 + 75.32^2] - C.F. = 36.020$$

ANALYSIS OF VARIANCE

Source	D.F.	S.S	M.S	F_c	$F_{t(5\%)}$
Row(Press)	2	283.761	141.88	17.34	
Column(Seq)	2	31.097	15.55	1.90	19.0
Pumps	2	36.020	18.01	2.20	
Error	2	16.360	8.18		

Sequence is not significant at the 5% level

Speed and Pressure w/ Sequence Averaged

Flow Degradation Ratio at 0-80

Press \ Speed	1500	2000	2500	
1500	0.952	0.966	0.978	2.896
2000	0.963	0.964	0.942	2.869
2500	0.920	0.927	0.948	2.795
	2.835	2.857	2.863	8.560

$$C.F. = \frac{(8.560)^2}{9} = 8.1415$$

$$TOTAL = (0.952)^2 + (0.966)^2 + \ldots + (0.927)^2 + (0.948)^2 - C.F. = .003$$

$$ROW\ (PRESS) = \frac{1}{3}[2.896^2 + 2.869^2 + 2.795^2] - C.F. = .0018$$

$$COLUMN\ (SPEED) = \frac{1}{3}[2.835^2 + 2.857^2 + 2.868] - C.F. = .0002$$

ANALYSIS OF VARIANCE

Source	D.F.	S.S	M.S	F_c	$F_{t(5\%)}$
Row(Press)	2	.0018	.0009	3.6	
Column(Speed)	2	.0002	.0001	0.4	6.94
Error	4	.0010	.00025		

Speed is not significant at the 5% level

Speed and Pressure with Sequence Averaged

Ferrographic Density Data at 0-80

Press \ Speed	1500	2000	2500	
1500	18.34	21.89	25.11	65.34
2000	28.64	25.53	25.99	80.16
2500	31.06	35.20	41.38	107.64
	78.04	82.62	92.48	253.14

$$C.F. = \frac{(253.14)^2}{9} = 7119.984$$

$$TOTAL = 18.34^2 + 21.89^2 + \ldots + 35.20^2 + 41.38^2 - C.F. = 389.63$$

$$ROW(PRESS) = \frac{1}{3}[65.34^2 + 80.16^2 + 107.64^2] - C.F. = 307.12$$

$$COLUMN(SPEED) = \frac{1}{3}[78.04^2 + 82.62^2 + 92.48^2] - C.F. = 36.32$$

ANALYSIS OF VARIANCE

Source	D.F.	S.S.	M.S.	F_c	$F_{t(5\%)}$
Row(Press)	2	307.12	153.56	13.29	6.94
Column(Speed)	2	36.32	18.16	1.57	6.94
Error	4	46.19	11.55		

Speed is not significant at the 5% level

Pressure with Sequence and Speed Averaged

Ferrographic Density Data at 0-80

1500 PSI	2000 PSI	2500 PSI	
18.34	28.64	31.06	
21.89	25.53	35.20	
25.11	25.99	41.38	
65.34	80.16	107.64	253.14

C.F. = 7119.984

TOTAL = 389.63

TRTSS(PRESS) = 307.12

ANALYSIS OF VARIANCE

Source	D.F.	S.S.	M.S.	F_c	$F_{t(5\%)}$
Pressure	2	307.12	153.56	11.16	5.14
Error	6	82.51	13.75		

Pressure is not significant at the 5% level

Pressure With Sequence and Speed Averaged

Flow Degradation Ratio at 0-80

1500 PSI	2000 PSI	2500 PSI	
0.952	0.963	0.920	
0.966	0.964	0.927	
0.978	0.942	0.948	
2.896	2.869	2.795	8.560

C.F. = 8.14151

TOTAL = .00288

$$TRTSS(PRESS) = \frac{1}{3}[2.896^2 + 2.869^2 + 2.795^2] - C.F. = .001834$$

ANALYSIS OF VARIANCE

SOURCE	D.F.	S.S.	M.S.	F_c	$F_{t(5\%)}$
PRESSURE	2	.001834	.00092	5.41	5.14
ERROR	6	.001050	.00017		

Pressure is not significant at the 5% level

SECTION VII

ELECTROSTATIC CHARGE EFFECTS ON FILTRATION PERFORMANCE

PROJECT PERSONNEL

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PREFACE

This section is concerned with the effects of electrostatic charge upon the performance of hydraulic fibrous filters. Specific objectives were the investigation of the charge generation and accumulation characteristics of a recirculating filtered hydraulic system. In addition, the effects of this charge upon filtration performance parameters was to be examined.

The experimental analysis included the verification of the generalized charge accumulation model by conducting tests at various fluid equilibrium conductivities and flow rates while taking measurements of the fluid charge density both upstream and downstream of the filter. The relationships between conductivity, particulate impurities, relative humidity and fluid additives and fluid types were also experimentally determined. Finally, the influence of electrostatic charge upon measurable filtration performance parameters was determined by conducting a number of multi-pass tests on

six different filter element models at various charge levels.

The experimental evaluations of the influence of electrostatic charge on filter performance resulted in definite relationships. Increasing charge levels and increasing particle sizes both resulted in an increase in the filter particle retention capabilities. The geometric mean diameter from the log-normal efficiency model decreased exponentially with increasing charge density and the geometric standard deviation showed a slight increase.

Recommendations are made for modifying the *"standard"* multi-pass filter test method by including the addition of a conductivity additive to the test system fluid before performing a test. This will increase the conductivity and thus reduce the charge influence below significant levels.

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CHAPTER I

INTRODUCTION

The degree of filtration required to protect current hydraulic systems is becoming more and more critical. Higher system pressure and greater complexity both demand more protection from abrasive contaminants. Excluding catastrophic failure due to fatigue, over-pressurizations, etc., the service life of most hydraulic components can be related directly to the amount of contamination entrained in the circulating fluid.

From a hydraulic system designers viewpoint the filter requirement can be simply stated as a filter which will provide adequate protection from particulate contaminants for a designated length of time with minimum pressure differential or flow restriction. From the filter manufacturers viewpoint this means that continual improvements must be made in the filter's particle separation characteristics as well as the contaminant capacity and pressure drop versus flow performance. It, therefore, becomes increasingly important to study the effects of all operating and design parameters on the performance of hydraulic filters for current at well as future applications.

It is well known that electrostatic forces play a significant role in the filtration of aerosols. In fact, the electrostatic precipitator relies entirely upon this phenomenonon. Only recently, however, has it been realized that under certain conditions, electrostatic

forces can significantly alter the separation performance of a hydraulic fluid filter.

The passage of hydrocarbons through microporous media (filters) can result in the generation of static electricity. This is attributed to the presence of ions in the fluid which separate into positive and negative charges. Since hydrocarbon liquids are generally poor conductors of electricity, such charges tend to accumulate as long as fluid movement continues. The flow of electricity produced by the capture (or loss) of electrons is commonly called streaming current. This streaming current and resulting charge has been recognized in fuel filtration systems as the cause of many accidental explosions due to electrical discharges in the atmosphere [1, 2].

The electrostatic charge which may build up in a filtered hydraulic system may have a definite effect upon the ability of the filter to separate and retain particulate contaminant. Conditions encountered in many fibrous filters are such that electrostatic forces may be a dominant factor in the removal of particles especially for particle sizes in the range at and below the effective filter pore size. The charge generation rate for fuels has been shown to be a function of the fluid properties, filter properties and flow rate.

Static electricity generation in hydraulic systems has been visually recognized at the Fluid Power Research Center either by sparks from or within a filter housing or a drop of oil which falls in an arc toward the filter housing. This phenomenon has generally occurred

when testing fine filter elements with housing bodies made of plastic or electrical insultating materials. However, recent tests witnessed at various laboratories indicated that charging may occur under a number of conditions especially with low relative humidity in the surrounding environment. The results from these tests certainly implied that this static charge has a definite influence upon the separation characteristics of a filter.

Because of these discoveries, this effort was undertaken to investigate the static charge phenomenon and its influence upon the filtration of hydraulic fluids. In specific, the areas investigated included static charge generation and charge accumulation in a hydraulic system environment. A generalized mathematical model for predicting electrostatic charge accumulation and relaxation in a hydraulic system was developed and verified. In addition, models for the particle separation performance characteristics of hydraulic filters including the influence of static charge were also developed and empirically verified.

The remainder of this report presents and discusses the results of the total research program. The following chapters review previous related investigations in the areas of static charge generation, charge relaxation, and filtration performance modeling. Chapter III presents a summary of mathematical models for predicting static charge accumulation and filtration performance as a function of charge level and other parameters. Chapters IV and V delineate the results of experimental programs to verify the charge accumulation and filter performance models, respectively. Finally, the remaining Chapters VI and VII discuss

applications of the research to current problems and present a summary and conclusions for the research effort. The Appendices contain detailed descriptions of the test facilities and procedures as well as a selected experimental data.

CHAPTER II

RELATED INVESTIGATIONS

An extensive literature survey has revealed that a number of investigations have been conducted relative to static charge generation in fuel systems. These studies have generally been concerned with charge accumulation from a safety standpoint. Only a few technical works have been published relative to the charge resulting from the flow of a hydrocarbon fluid through a micro-porous filter. None of the investigations found, however, have been concerned with hydraulic fluids. The literature is almost completely void of investigations relating to static charge effects on particle capture and retention in hydrocarbon fluids.

ELECTROSTATIC CHARGE GENERATION

The charging process in a hydrocarbon fluid system actually begins before the fluid is in motion. At the interface between the liquid and the solid boundary, an uneven distribution of ions is observed. The solid phase (in this case, the pipe wall or filter pore surface) will carry a net charge of one polarity and the liquid phase will contain an equal but opposite charge. This is the concept of the electric double layer [1]. A strictly non-ionic liquid would have the properties of an insulator. Ultra-pure

hydrocarbons approach this category, however, even minute traces of contaminants can significantly alter this property.

The streaming current is generated as the fluid flows past the solid boundary picking up electrical charge from the double layer area. Since filters present a tremendous amount of surface area in which this charge separation can occur, they constitute a major source of electrostatic charge generation. Klinkenberg [1] utilized the classic work of Helmholz to evaluate the streaming current in an insulating tube. This development assumes that the double layer is thin relative to the pipe diameter and thus is not particularly suited for prediction of streaming current generated in microcapillaries. Other investigators [3, 4] have studied flow in microcapillaries and have removed the restriction concerning the relatively small double layer thickness. These authors, however, have not considered the problem of low fluid conductivity or tortuous passage hydrodynamics.

The conductivity, K, of the fluid (the inverse of resistivity) is a measure of the discharge time of a capacitor which uses the fluid as a dielectric. The units of conductivity are $\frac{1}{\text{Ohm m}}$ or $\frac{\text{mho}}{\text{m}}$. A very pure hydrocarbon may have a conductivity of $10^{-3} > K > 10^{-13}$ mho/m; superpure $10^{-15} >$ however, a fluid with ionic additives would be expected to yield a much larger value. An often used

quantity which depends most heavily upon fluid conductivity is relaxation time, τ, which is defined as:

$$\tau = \frac{\varepsilon \varepsilon_o}{K} \qquad (2\text{-}1)$$

where:

ε = dielectric constant of the liquid

ε_o = absolute dielectric constant of a vacuum

One of the first reported studies dealing with charge generation during hydrocarbon flow through a microporous filter was reported by Gavis and Wagner (5) in 1968. They conducted a number of experiments then developed expressions empirically, based upon ideas drawn from theory. They utilized n-heptane fluid doped with various amounts of an ionizing additive to control the conductivity. The test filters were various types of Millipore membrane disks. Their test results indicated the polarity of the charge generated was a function of the mechanical properties. The describing relationship they developed can be expressed as:

$$I_s = 2 \times 10^{-4} \, d \, A_f \, PF \, C_o \left(\frac{v\tau}{d}\right)^{1.75} \qquad (2\text{-}2)$$

where:

I_s = streaming current, amp

d = nominal pore diameter, m

A_f = filter area, m^2

P = filter porosity (void volume/total volume)

F = Faraday constant

C_o = original total ion concentration in liquid, mol/m^3

v = superficial flow velocity, no volume flow rate per total cross-sectional area, m/sec

for $\frac{v\tau}{d} < 5 \times 10^{-3}$.

The term, $v\tau/d$, rewritten as $v/(d/\tau)$ is the ratio of the convective velocity to conductive velocity of ionic movement. They determined C_o experimentally from the conductivity, K, and an empirical estimate of ion mobility. They concluded that current generated was independent of the ratio of pore length to diameter and that the charging rate should be independent of the structure of the microporous medium.

Another pertinent study was conducted by Leonard and Carhart [6] on the effect of conductivity on charge generation in hydrocarbon fuels flowing through fiber glass filters. They utilized JP-4 and JP-5 jet fuel with various amounts of conductivity improver additives. Their results showed that the magnitude of the streaming currents developed increased to a maximum then decreased as conductivity was increased. Experimentally, they determined that for a given filter and fluid, the current developed increased with increasing flow rate. This is in agreement with the work previously conducted by Gavis and Wagner [5].

Lauer and Antal [7] conducted in investigation of charge generation of several hydrocarbons flowing through low conductivity filter media. They found for zylene flowing through teflon media

that streaming current was found to be linearily proportional to flow velocity and increased with decreasing pore size. Among other things, they confirmed some of the previous conclusions of Gavis and Wagner [5].

Probably the most pertinent study to date was conducted by Huber and Sonin [8] in which they considered the flow of n-heptane doped with anti-static additive through microporous disk filters. Their experiments were conducted in a similar manner to those by Gavis and Wagner [5]; however, their theory was derived in a generalized nature. They concluded [9] that the equations developed by Gavis and Wagner [5] would not be interpreted as having general validity and that the correlation established between streaming current, filter pore radius and fluid conductivity is in error. Huber and Sonin's theory [8] considered a fixed charge which is absorbed on the interior surfaces of the porous filter. Previous investigators has omitted this phenomena. Huber and Sonin's governing equation for high fixed charge density [8] is the following:

$$I_s = - \frac{Z_f A_f}{\ell_o} \left[\frac{T^2}{P} \frac{v h K}{u} + F(t_a) \frac{\varepsilon \varepsilon_o v^2}{u} \right] \qquad (2\text{-}3)$$

where:

Z_f = sign of filter's fixed charge (± 1)

ℓ_o = equivalent electrode separation, $\ell_o \equiv K A_f R_o$, m

T = tortuosity of filter pores

h = filter medium thickness, m

u = mobility of ion (with sign opposite to that of filters fixed charge) m^2/volt sec.

$F(t_a)$ = dimensionless function of t_a given by Eq. (17) of [8].

t_a = transport number of that ion species in fluid which has charge of polarity opposite to that of the fixed charge associated with the filter.

R_o = total electrical resistance of fluid between electrodes.

Huber and Sonin's experiments [10] as well as those conducted by Gavis and Wagner [5] correlate extremely well with the theory of Huber and Sonin [8]. An important result of Huber and Sonin's studies [8] is that the streaming current is not dependent upon filter pore size nor on fluid conductivity K alone; but I_s depends upon the equivalent electrode spacing which includes the resistance of the fluid between the measuring electrodes.

CHARGE RELAXATION

Because of the recirculating nature of hydraulic systems, it is necessary to consider the relaxation or dissipation characteristics of electrostatic charge in hydraulic fluids. In general, the charge that has been entrained in a liquid will dissipate when the source of the charge is removed. The relaxation of charge for a stationary fluid was reviewed by such investigators as Klinkenberg [1], Winter [11], and Agnew [12] who concluded that the relaxation is governed by an exponential law that is dependent upon the initial fluid charge and the fluid relaxation time constant. Vallenga and

Klinkenberg [13] and Carruthers and Marsh [14] continued the analysis for a fluid flowing through pipes.

For a fluid at rest, all investigators mentioned above conclude that the charge decay can be described by the following equation:

$$\frac{ds}{dt} = -\frac{s}{\tau} \qquad (2\text{-}4)$$

where:

s = electrical charge density per unit fluid volume, C/m^3

τ = fluid relaxation time, sec. = $\varepsilon\ \varepsilon_o/K$

Eq. (2-4) is based upon the assumption that transport of charge by diffusion or convection can be ignored. The equation is derived from Ohm's Law which requires the above assumption. Diffusion and convection should be included for an accurate description; however, diffusion effects only become important in small laboratory apparatus or very near an interface where large concentration gradients occur so that in full scale equipment the contribution of diffusion can be ignored [15]. For a fluid in motion, convection effects can be eliminated by letting the elemental volume of charge "ds" move along with the liquid [13]. This corresponds to the Lagrangian description of motion in hydrodynamics. Eq. (2-4) then becomes the governing relationship for the discharge or relaxation of electrically charged hydrocarbon liquids.

Winter [11] and later Vallenga and Klinkenberg [13] discuss two different types of fluid conductivity--equilibrium and effective.

The equilibrium conductivity is the conductivity measured in a fluid at rest or under steady-state conditions (e.g., one moving in a long straight path) when it is under no significant electrical unblanace. The effective of "in situ" conductivity corresponds to the rate of relaxation of charge in a fluid at any charged condition to which it is subjected.

Exponential Law of Charge Relaxation

For a hydrocarbon liquid with relatively high equilibrium conductivity (magnitude much higher than a pure hydrocarbon), it is normal and generally permissible to ignore the difference between effective and equilibrium conductivity. Essentially, then, the conductivity can be treated as a constant for a high conductivity liquid. Thus, integration of Eq. (2-4) yields:

$$s = s_o \, e^{-t/\tau} \tag{2-5}$$

for τ = constant and s_o the initial charge density. Eq. (2-5) is generally used to describe the relaxation of a charged hydrocarbon liquid with constant electrical conductivity.

If the liquid is in motion such as flow in a pipe, Carruthers and Marsh [14] conclude that a total description of the relaxation must include the charge which is generated at the liquid/wall interface during passage through the pipe. They developed, from the concepts in [1], an equation for the current carried by the liquid as

$$I_t = I_o \, e^{-t/\tau} + I_i(1 - e^{-t/\tau}) \tag{2-6}$$

where:

I_t = current carried in the liquid at time t, amp

I_i = Steady current generated at the liquid/wall interface, amp

I_o = Current carried in the liquid at time = o, amp

The first term in Eq. (2-6) represents the exponential law of relaxation while the second term is the current generated by the flow of the liquid. Eq. (2-6) was developed for pipe with a high electrical conductivity such as is generally the case in hydraulic test systems. The value of I_i can be calculated by methods presented by Klinkenberg [1].

Hyperbolic Law of Charge Relaxation

Vallenga and Klinkenberg [13] present a different theory for charge relaxation that does not assume constant conductivity. The effective conductivity they conclude is a linear function of the charge density or:

$$K = CFu \neq \text{constant}$$

where:

C = ion concentration, mol/m^3

u = ion mobility, $m^2/volt\ sec.$

Since the charge density $s = CF$ and $\tau = \varepsilon\ \varepsilon_o/K$, the equivalent of Eq. (2-4) for non-constant K is:

$$\frac{ds}{dt} = -\frac{u\ s^2}{\varepsilon\ \varepsilon_o} \qquad (2\text{-}7)$$

which can be integrated to give:

$$\frac{1}{s} = \frac{u\,t}{\varepsilon\,\varepsilon_o} + \frac{1}{s_o} = \frac{t/\tau_o + 1}{s_o} \qquad (2\text{-}8)$$

where τ_o is the relaxation time and s_o is the initial charge density both at time = o.

Equation (2-8) is the hyperbolic law of charge relaxation as derived by Vallenga and Klinkenberg [13] describes the rate of discharge of an electrically charged liquid if the conductivity is directly proportional to the charge density.

CHARGE EFFECTS ON PARTICLE CAPTURE

The important mechanisms for particle separation in a fibrous filter other than sieving or mechanical blockage are generally classified as:

1. Direct interception
2. Inertial impaction
3. Browman diffusion
4. Gravity settling
5. Electrostatic precipitation

Fig. 2-1 illustrates these mechanisms for flow past a typical filter fiber. Electrostatic attraction can occur as a result of electrostatic forces drawing a particle from the fluid to the fiber surface. As shown in Fig. 2-1, it is possible to collect particles by electrostatic means when the particle is so far from the fiber that other mechanisms such as direct or inertial interception would be ineffective. If the electrostatic forces are large enough to draw a par-

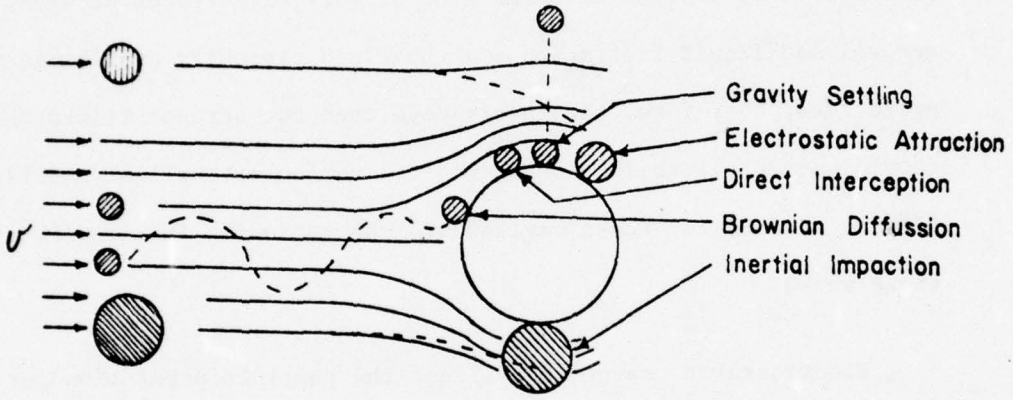


Figure 2-1. Typical Particle Capture Mechanisms for Fibrous Filters

ticle from the flow stream, then this capture mechanism may become quite significant.

A thorough survey of the literature revealed no previous investigations of the effects of electrostatic charge on filtration by fibrous hydraulic filters. A number of investigators [16-21], however, have studied the effect of electric forces on the filtration of aerosols by fibrous filters. The primary differences between aerosol and liquid filtration are the fluid viscosity and chemical properties; therefore, the theory developed for aerosol filtration should have applicability to liquid filtration with slight modification to account for these variables. The following is a review of those works.

Electrostatic charges may affect the particle separation performance of a filter in two ways: (1) particles may be attracted to the fibers from a distance, and (2) the electrostatic forces may increase the ability of a fiber to retain a particle once they have come in contact. The experiments of Loffler [22], however, have shown for aerosols that these forces are not dominant adhesive forces once a particle has been captured. His data indicated that Van der Wals forces were the important adhesive forces. This should be even more true for liquid filters where the higher viscosities would tend to detach the particles from a fiber with even greater drag force. Thus, this study is only concerned with the effects of the electrostatic capture mechanism.

Kraemer and Johnstone [16] investigated the electrostatic collection of particles by a spherical collector. In doing so they classified the electrical forces into five categories as follows:

1. Coulombic forces between a charge particle and a charged collector.
2. Force between a charged particle and its image on an uncharged collector.
3. Force between a charged particle and its image on an uncharged aerosol particle.
4. Force of repulsion on particle being collected by like charge on surrounding particles.
5. Force between a charged particle and the total charge induced in the collector by surrounding charged aerosol.

For the case of hydraulic filtration where the fluid carries a charge due to "election stripping" in the filter it is reasonable to assume that the filter fibers will be charged. It is possible that the contaminant particles may or may not be charged depending upon whether or not the particles take on a charge from the fluid. If the electrostatic charge accumulates in the recirculating system, then it is possible for the particles to be charged with the same polarity as the fluid (opposite of the filter fibers); however, if the charge in the fluid is relaxed before it circulates around the system, the particles would most likely not be charged. Because of this situation, it would be possible that either cases 1 or 3 above (charged

or uncharged particles) might be applicable to hydraulic filters.

Kraemer and Johnstone [16] developed equations for cases 1 and 3 above for both spherical and cylindrical collectors. Their expression for single fiber efficiency of a cylindrical collector for case 1 with both collector and particle charged was:

$$\eta_{s1} = \frac{C_x \, q_c \, q_p}{3 \mu \, \varepsilon_o \, D_p \, v} \qquad (2\text{-}9)$$

where:

η_{s1} = single fiber efficiency for case 1 above

C_x = Cunningham correction factor

q_c = charge on collector per unit area, C/m^2

q_p = charge on particle, C

μ = fluid viscosity, cp = .01 dyne sec/cm^2

D_p = particle diameter, m

v = fluid velocity, m/sec

For case 3 above with charged collector and uncharge particles, their equation was:

$$\eta_{s3} = \left[\frac{\pi(\varepsilon_p - 1) \, C_x \, q_c^2 \, D_p^2}{2(\varepsilon_p + 2) \, \mu \, D_f \, v_o \, \varepsilon_o} \right] 0.33 \qquad (2\text{-}10)$$

where:

η_{s3} = single fiber efficiency for case 3 above

ε_p = dielectric constant of particle

D_p = fiber diameter, m

Single fiber efficiency is generally defined as the ratio of the flow stream area from which all particles are removed to the projected

fiber area, both areas taken perpendicular to the direction of free stream flow. Figure 2-2 illustrates the relationship for single fiber efficiency with the limiting trajectory exaggerated for electrostatic influences. The single fiber efficiency, η_s, is then defined by the following equation.

$$\eta_s = \frac{2\, y_o}{D_f} \tag{2-11}$$

where y_o is the ordinate of the limiting particle trajectory in the undisturbed flow in front of the filter fiber and D_f is the fiber projected diameter. The single fiber efficiency, thus, is basically an indication of the ability of a filter fiber to capture particles from the flow stream around it. As illustrated in Fig. 2-2, when electrostatic forces are involved, single fiber efficiency may easily be greater than unity.

If we let the constants Θ_1 and Θ_2 be defined as follows:

$$\Theta_1 \equiv \frac{C_x}{3\mu\, \varepsilon_o\, v} \tag{2-12}$$

$$\Theta_2 \equiv \left[\frac{\pi\, (\varepsilon_p - 1)\, C_x}{2(\varepsilon_p + 2)\mu\, D_f\, v_o\, \varepsilon_o}\right]^{0.33} \tag{2-13}$$

then Eqs. (2-9) and (2-10) can be rewritten as

$$\eta_{s1} = \frac{\Theta_1\, q_c\, q_p}{D_p} \tag{2-14}$$

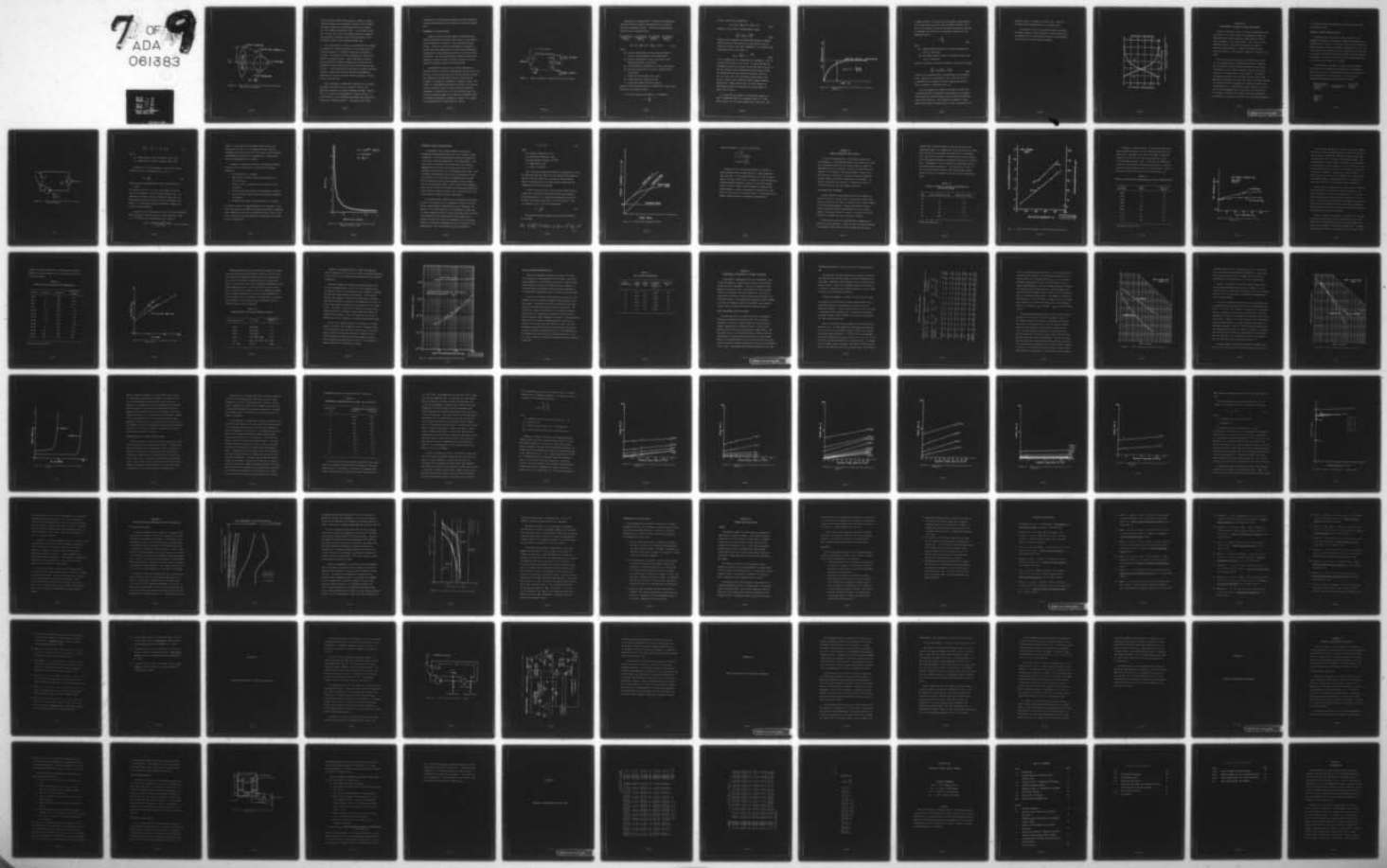
$$\eta_{s3} = \Theta_2\, q_c^{.66}\, D_p^{.66} \tag{2-15}$$

Both Eqs. (2-14) and (2-15) predict that the single fiber efficiency

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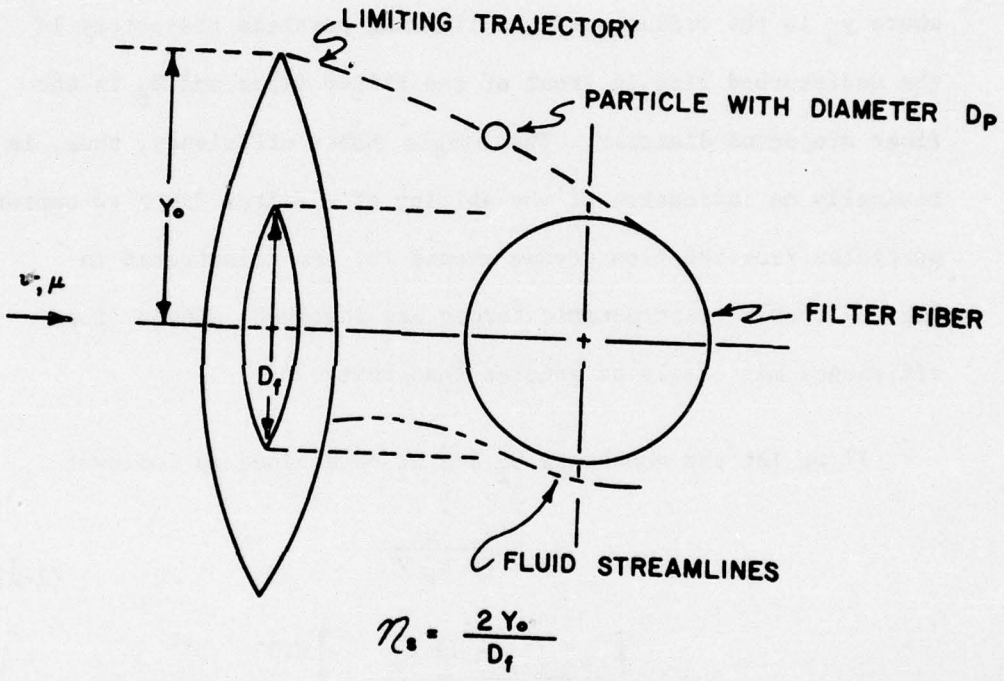


Figure 2-2. Model of Particle Trajectory Near Single Fiber With Electrostatic Influence

should increase as charge levels increase. However, a primary difference between the two equations is that Eq. (2-14) predicts decreasing efficiency with increasing particle size; whereas, Eq. (2-15) predicts the opposite effect. It will later be shown that the form of Eq. (2-15) for uncharged particles is generally more applicable to hydraulic system filtration because of the increasing efficiency with increasing particle size.

For a higher degree of accuracy, Loffler and Muhr [21] suggest the addition of an adhesion probability to the single fiber efficiency expressions. The adhesion probability they define as the ratio of the number of particles adhering to a fiber to the number of colliding particles. Earlier experiments by Loffler were mentioned where adhesion probabilities less than unit were determined for solid particles with fluid velocities greater than 25 cm/sec. Loffler and Muhr [21] found that the collision efficiency (product of single fiber efficiency and adhesion probability) depends upon the particle and fluid physical properties, the flow velocity, and fiber size.

Direct measurement of single fiber efficiency or of collision efficiency as defined by [21] as a function of each of the significant variables, is an ideal fundamental approach. However, this is not practical for measurements on complete filters. If a viscous liquid is the surrounding fluid medium, this measurement task becomes increasing difficult. The approach taken by many

investigators is to measure experimentally the overall separation efficiency then develop empirical equations to describe the phenomena.

MEASUREMENT OF FILTER EFFICIENCY

A number of studies have been conducted at Oklahoma State Univeristy during the past several years relative to the measurement and description of hydraulic filter separation efficiency [25-26]. Tucker [23] in his Ph.D. dissertation considered hydraulic fluid flow through wire cloth filter media and developed expressions for filter efficiency based upon the filter pore size distribution. The equations developed by Tucker are not directly applicable to fibrous or depth type filters because of the randomness and complicated nature of the pore structure.

A standardized and realistic test has been developed for hydraulic filters as a result of investigations reported in [24, 25]. This test, called the multi-pass test, has since become a national and international standard method [27] and is being utilized worldwide to determine the performance of hydraulic filters. The current U.S. Army MERADCOM specification is based on the technique. The multi-pass test basically involves injecting a specified contaminant, AC Fine Test Dust, into the circulating filter test system and extracting samples of the influent and effluent fluids to determine the separation performance. Figure 2-3 is a simplified representation of the multi-pass test circuit.

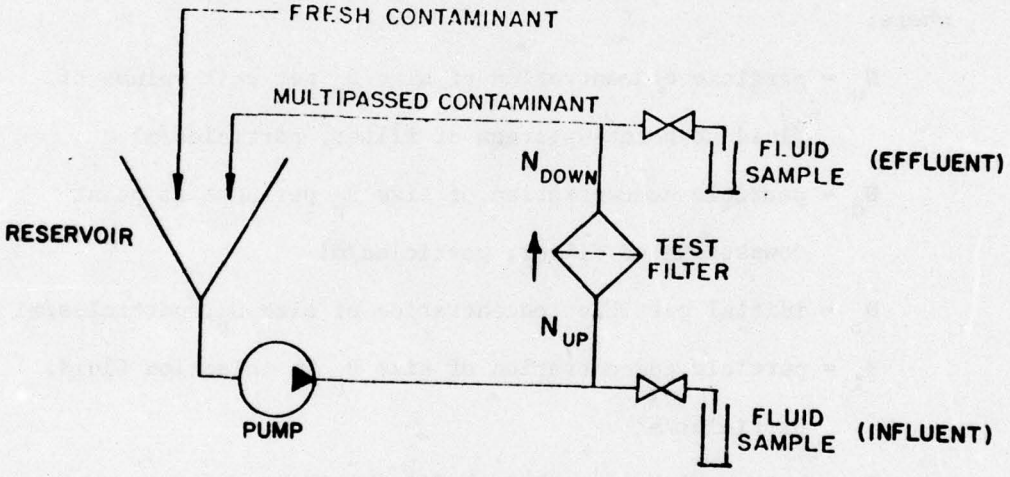


Figure 2-3. Simplified Schematic of Multi-Pass Filter Test Circuit

Equations can be developed [24] to describe the contamination level and effective filtration characteristics as a function of time and the operating variables. The basic governing equation can be written in integral form as:

No. Particles in Reservoir of Size D_p	=	No. Particles Originally of Size D_p	+	No. Particles Injected of Size D_p	-	No. Particles Removed of Size D_p

$$N_u V = N_o V + \int N_i Q_i \, dt - \int (N_u - N_d) Q \, dt \qquad (2\text{-}16)$$

where:

N_u = particle concentration of size D_p per unit volume of fluid at point upstream of filter, particles/ml

N_d = particle concentration of size D_p per unit at point downstream of filter, particles/ml

N_o = initial particle concentration of size D_p, particles/ml

N_i = particle concentration of size D_p in injection fluid, particles/ml

Q = volume flow rate through filter, ℓ/min

Q_i = volume flow rate of injection, ℓ/min

V = circulating volume of fluid in system, ℓ

Equation (2-16) describes the particle concentration of a given size particle D_p in the upstream fluid.

If the filter separation efficiency, η, is defined as:

$$\eta = 1 - \frac{N_d}{N_u},$$

Then Eq. (2-16) can be rewritten as:

$$N_u V = N_o V + \int N_i\, Q\, dt - \int \eta\, N_u\, Q\, dt \qquad (2\text{-}17)$$

Equation (2-17) can then be differentiated to give:

$$\frac{d\, N_u}{dt} + \frac{\eta\, Q}{V} N_u = \frac{N_i\, Q_i}{V} \qquad (2\text{-}18)$$

Equation (2-18) represents the controlling differential equation which describes the concentration of a given particle size upstream of the filter at any time, regardless of the injection rate. Integration of Eq. (2-18) results in

$$N_u(t) = \frac{N_i\, Q_i}{\eta\, Q}\left(1 - e^{-\frac{\eta Q}{V} t}\right) \qquad (2\text{-}19)$$

if it is assumed that η is constant and N_o is negligible. Fig. 2-4 illustrates the solution to Eq. (2-19). It can be seen that the contamination level will stabilize at some level equal to $N_i\, Q_i/\eta\, Q$. The time constant for the stabilization period is given by $V/\eta Q$. The standard multi-pass test method [27] requires a ratio of $V/Q = 0.25$ min.; thus, the time constant becomes simply $0.25/\eta$ minutes. Because a filter generally exhibits higher separation efficiencies at higher particle sizes, the time required for stabilization of particle concentration is usually higher for smaller particle sizes.

It is generally more common in the Fluid Power Industry to report contamination levels on a cumulative basis, or in other words, particles per unit volume greater than a given size. Thus,

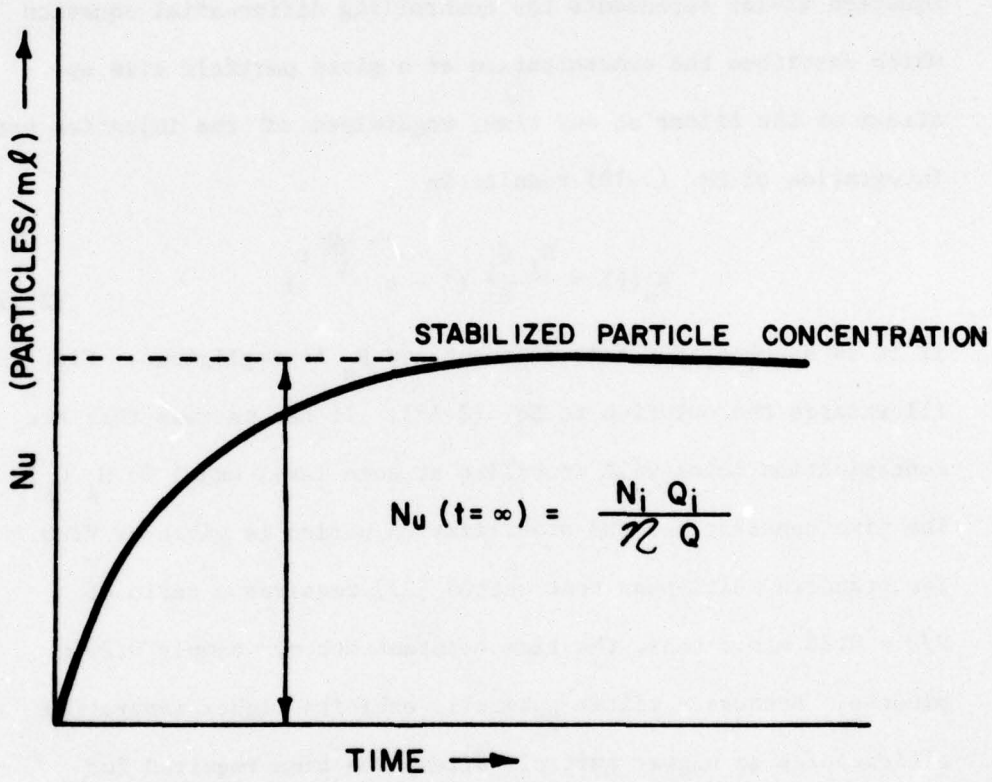


Figure 2-4. Graphical Representation of Multi-Pass Filtration Equation

a common term used to describe the filter separation characteristics is the filtration ratio or beta ratio (β) which is defined as the ratio of the cumulative particle concentration greater than some size D_p upstream of the filter to the respective concentration in the downstream fluid or:

$$\beta = \frac{\overline{N}_u}{\overline{N}_d} \tag{2-20}$$

where:

$\overline{N}_u$ = upstream cumulative particle concentration greater than size D_p, particles/ml

$\overline{N}_d$ = downstream cumulative particle concentration greater than size D_p, particle/ml

Equation (2-18) can be rewritten in cumulative terms as the following:

$$\frac{d\,\overline{N}_u}{dt} + \frac{(\beta - 1)Q}{\beta V}\,\overline{N}_u = \frac{\overline{N}_i\, Q_i}{V} \tag{2-21}$$

where $\overline{N}_i$ is the cumulative particle concentration of the injection fluid. Because of the similarities of Eqs. (2-18) and (2-21) it is expected that the cumulative particle concentration would behave in a manner as illustrated in Fig. 2-4.

Fitch and Tessmann [26] extended the concepts of Tucker [23] to graphically model the separation characteristics of a hydraulic filter based upon cumulative particle size distributions resulting from the multi-pass test. They considered the percent of contaminant which permeates through a filter to result in penetration and

retention curves as a function of particle size. Figure 2-5 illustrates these characteristics for a typical filter.

This study will utilize the multi-pass filter test method and concepts similar to those presented by Tucker [23] and Fitch and Tessmann [26] for illustrating the change in the performance characteristics with electrostatic influences.

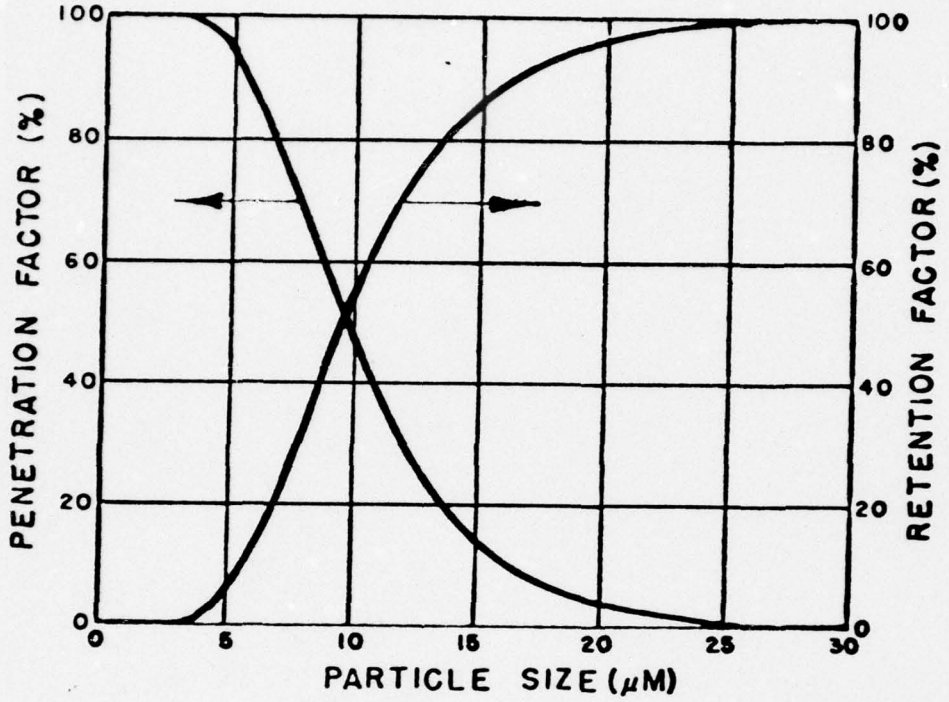


Figure 2-5. Filter Element Performance Characteristics Expressed on a Cumulative Basis

CHAPTER III

DEVELOPMENT OF CHARGE ACCUMULATION MODELS

Chapter II presented a review of previous investigations which were related to the present study. Relationships were shown for charge generation, charge relaxation, charge effects on aerosol filter performance, and filtration models. This chapter presents mathematical models to describe such parameters in terms of hydraulic system filtration. A complete development of these models are presented in [28] and only a summary is included in this report.

The flow of hydraulic fluid in a typical fluid power system is not representative of the fueling process from the standpoint that the hydraulic fluid is continuously being recirculated while fuel generally passes through a system one time into a receiving vessel such as storage tank or vehicle reservoir. All previous investigations reported in the literature have dealt with charge generation and relaxation in a fuel type system. The theory developed is therefore not directly applicable to a hydraulic system because charge generated by a liquid passing through a hydraulic filter may not be fully discharged before it again enters the filter during recirculation. Based upon the theories presented by Vallenga and Klingenberg [13], descriptive equations: for the charge accumulation

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in a hydraulic system were developed in [28] for both exponential and hyperbolic decay.

EXPONENTIAL CHARGE ACCUMULATION MODEL

According to most investigators cited in Chapter II, the discharge or relaxation of eletrostatic charge can be modeled by an exponential law. In order to relate this law in terms of a multi-pass hydraulic system, consider the schematic illustrated in Fig. 3-1. For the purposes of this derivation it will be assumed that the pipe wall and reservoir are electrically conductive (metal) and that the system is grounded. In addition, the flow rate Q and the charge generated by the filter s_f are assumed to be constant.

Since the system is grounded, the charge in each incremental fluid volume can be expressed in accordance with the exponential law given by Eq. [2-4]. A differential equation can be written for the charge density in the fluid at the out of the reservoir (assuming instantaneous and uniform mixing) as follows:

Rate of change in charge density at outlet of reservoir = − Exponential decay in reservoir − Rate leaving outlet of reservoir +

Charge rate added to reservoir inlet

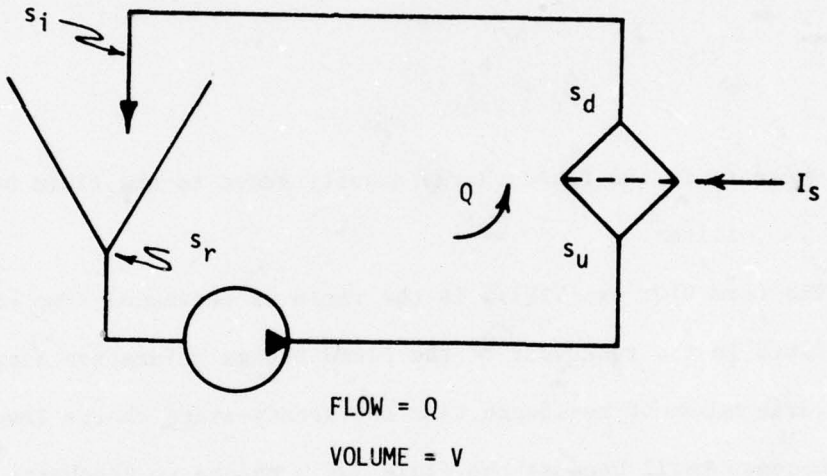


Figure 3-1. Charge Accumulation Schematic of Multi-Pass Hydraulic System

$$\frac{d\,s_r}{dt} = -\frac{s_r}{\tau} - s_r\,\frac{Q}{V} + s_i\,\frac{Q}{V} \qquad (3\text{-}1)$$

where:

s_r = charge density of fluid at reservoir outlet, C/m^3

s_i = charge density of fluid at reservoir inlet, C/m^3

Equation [3-1] can be manipulated to derive the following expression for $\hat{s}_r$, the limiting value of s_r as $t \to \infty$:

$$\hat{s}_r = \frac{s_f}{V/Q\tau} \qquad (3\text{-}2)$$

where, s_f is the fixed charge density added to the fluid by the filter.

The term $V/Q\tau$ or $(V/Q)/\tau$ is the ratio of residence time of the fluid in the reservoir to the fluid charge relaxation time. For large values of residence time the steady-state charge level, $\hat{s}_r$, becomes small because the fluid has a chance to discharge in the reservoir. If V/Q is small, however, the reservoir charge level can build to extreme amounts.

As an example of the relationship of Eq. [3-2], consider a hydraulic fluid with dielectric constant equal to 2.0 and conductivity equal to 1×10^{-12} mho/m; thus, the relaxation time is equal to

$$\tau = \frac{\varepsilon\,\varepsilon_o}{K} = \frac{(2)\ (8.85 \times 10^{-12}\ \text{sec}/\Omega\text{m})}{1 \times 10^{-12}/\Omega\text{m}} = 17.7\ \text{sec} = 0.3\ \text{min}$$

Figure 3-2 illustrates the relationship between $\hat{s}_r$ and s_f for various ratios of V/Q. For the standard multi-pass filtration performance test [27] the ratio V/Q = 0.25 minutes, thus the charge accumulating in the reservoir is approximately 1.2 times greater than the charge generated by the filter.

In summary the exponential equation for describing the charge buildup in a recirculating system is based upon the following assumptions:

1. Fluid conductivity is constant.
2. Mean velocity of charge carriers and mean velocity of liquid are equal.
3. Electric field is constant over cross-section of liquid and piping.
4. Uniform instantaneous fluid and charge mixing in reservoir.
5. Pipe and reservoir walls are metal (electrically conductive) and grounded.
6. No significant charge is generated outside of the filter.

For greater accuracy, the charge generated at the liquid/wall interface in the piping can be added to the equation for charge accumulation. Tests conducted as part of this investigation without a filter in the circuit showed that this charge generation was generally low for most situations encountered.

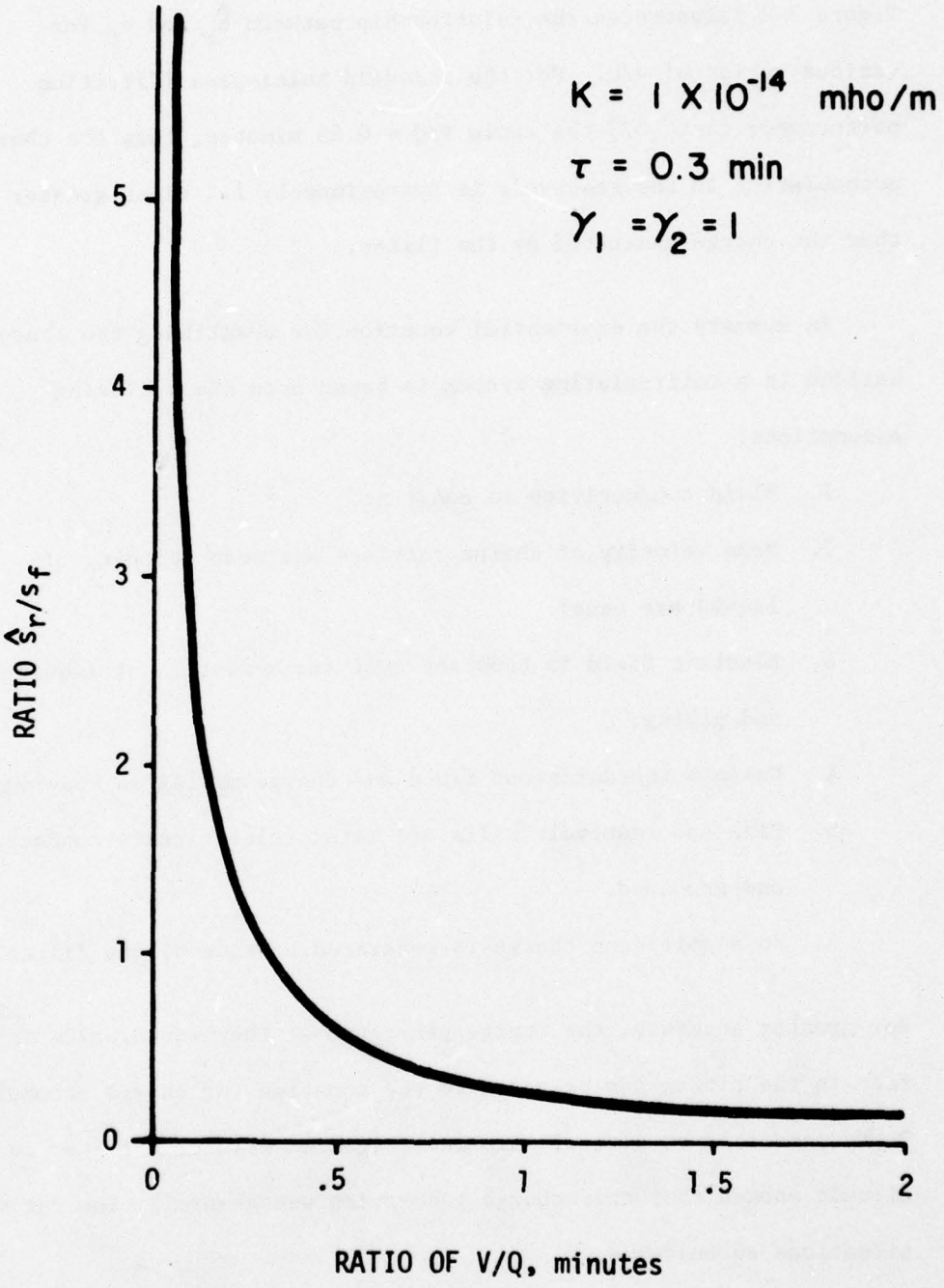


Figure 3-2. Relationship Between Charge Accumulation and Reservoir Residence Time

GENERALIZED CHARGE ACCUMULATION MODEL

The hyperbolic law of charge relaxation as presented by Vallenga and Klinkenberg [13] could also be used to predict charge accumulation. This rule applies when the electrical conductivity of the fluid is directly proportional to the charge density. This would occur, in an extreme case, when all ions of one polarity, depending upon the affinity that the particular filter has for absorbing a certain polarity ion, are removed from the liquid. Such a condition could generally exist only for fluids with an extreme case, when all ions of one polarity, depending upon the affinity that the particular filter has for absorbing a certain polarity ion, are removed from the liquid. Such a condition could generally exist only for fluids with an extremely low equilibrium conductivity and the effective conductivity would, thus, be entirely due to ions of the sign that constitute the net charge density.

The hyperbolic model requires that the equilibrium conductivity be equal to zero when there is no unbalance of charge in the fluid. Because common hydraulic fluids definitely have a measurable conductivity in an equilibrium state, it is reasonable to assume that the charge relaxation could not follow the hyperbolic model precisely. A more generalized and realistic model for the conductivity of hydraulic fluids can be written as a combination of an equilibrium conductivity term and an additional term that is a function of the charge density. Thus, the conductivity can be expressed as:

$$K = K_e + K_o s^n \tag{3-3}$$

where:

K = effective conductivity, $1/\Omega m$

K_e = equilibrium conductivity, $1/\Omega m$

K_o = proportionality constant, $m^{3n-1}/\Omega C^n$

s = charge density, C/m^3

n = constant in exponent

Such a conductivity generalized model has characteristics of both the exponential law were conductivity is constant and the hyperbolic law where effective conductivity increases as charge increases. Figure 3-3 illustrates the relationship between conductivity and charge density for the various models.

In order to utilize Eq. [3-3], the actual fluid charge density is normalized by a theoretical value which the fluid would possess at equilibrium conditions. By using the relationships from [13] this reference value would be K_e/u where u is the ion mobility. Thus, the constant K_o in Eq. [3-3] can be expressed as:

$$K_o = \frac{u^n}{K_e^{n-1}} \tag{3-4}$$

The generalized charge relaxation law was derived in [28] as the following:

$$\frac{d s_r}{dt} = - \frac{s_r}{\tau_e} - \frac{K_o s_r^{n+1}}{\epsilon \epsilon_o} - \frac{s_r Q}{V} + \frac{Q}{V}\left\{ Y_3\left[s_f + Y_4 s_r^{-n} + \frac{K_o}{K_e}(Y_4 - 1)\right]^{-\frac{1}{n}} + \frac{K_o}{K_e}(Y_3 - 1)\right\}^{-\frac{1}{n}} \tag{3-5}$$

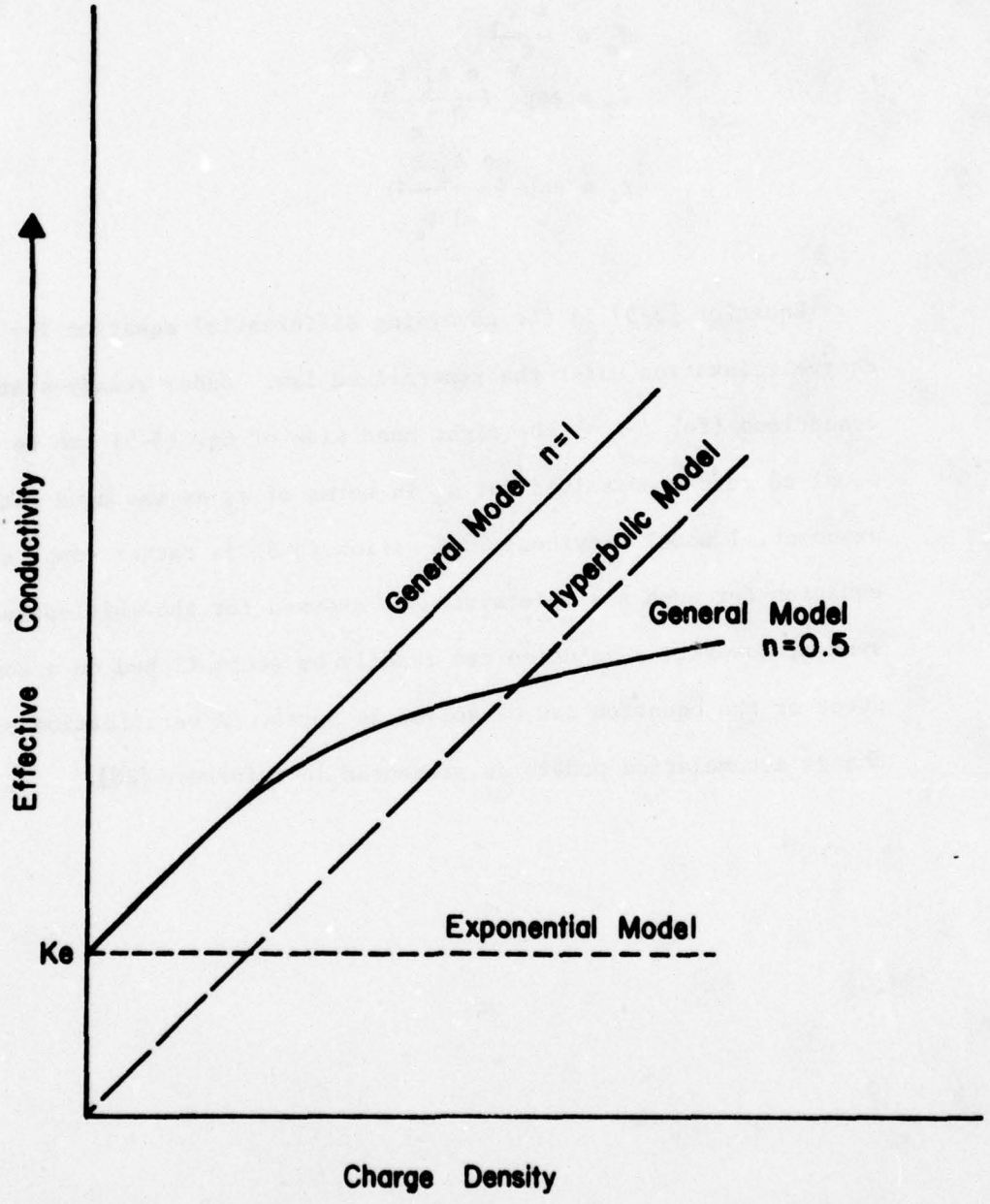


Figure 3-3. Various Fluid Conductivity Models

where the constants τ_e, γ_3 and γ_4 be defined as:

$$\tau_e \equiv \frac{\varepsilon\, \varepsilon_o}{K_e}$$

$$\gamma_3 \equiv \exp\left(\frac{n\, A_\ell\, \ell_d}{Q\, \tau_e}\right)$$

$$\gamma_4 \equiv \exp\left(\frac{n\, A_\ell\, \ell_u}{Q\, \tau_e}\right)$$

Equation [3-5] is the governing differential equation for charge relaxation under the generalized law. Under steady-state conditions (for $\tau \to \infty$) the right hand side of Eq. (3-5) can be set equal to zero then solved for $\hat{s}_r$ in terms of s_f as was done with the exponential model previously. Equation (3-5) is rather complex equation for such a simple system as assumed for the multi-pass system; however, a solution can readily be accomplished on a computer or the equation can be solved in parts. A verification of the change accumulation models is presented in reference [28].

CHAPTER IV

CHARGE GENERATION TEST RESULTS

In order to determine levels of electrostatic charge which can accumulate in a recirculating hydraulic system and also to verify the relationships presented in the previous chapter, a number of experimental tests were conducted. Because the fluid conductivity appeared to be an important controlling parameter, several tests were performed to determine the relationship between conductivity, relative humidity, particulate contamination, and additives. In addition, experiments were conducted to determine the effects of conductivity and flow rate upon charge accumulation.

FLUID CONDUCTIVITY EXPERIMENTS

Personal experiences and discussions with other laboratories have led the author to believe that the electrostatic charge accumulation in a hydraulic test stand is a function of the relative humidity of the surrounding atmosphere. The most obvious method by which relative humidity may affect charge generation is by an apparent change in the effective fluid conductivity. A series of tests were conducted to determine this effect of relative humidity.

The multi-pass filter test stand described in Appendix A was utilized in these experiments. Before starting, the stand was drained and thoroughly flushed, then new fluid was added directly from an

unopened drum. MIL-H-5606 hydraulic fluid was utilized in most experiments because it is commonly used in aircraft hydraulic systems and is specified in the standard multi-pass filter test procedure [27]. The fluid was allowed to circulate while the relative humidity of the surrounding environment was maintained constant for at least 48 hours. Measurements of fluid conductivity were then made using the DC method described in Appendix C. The concentration of water in the oil was also measured using the Karl Fischer reagent method. Table 4-1 and Fig. 4-1 delineate the results from this experimental phase.

TABLE 4-1

EFFECT OF RELATIVE HUMIDITY ON CONDUCTIVITY AND WATER CONTENT

RH%	Water Concentration, ppm	Conductivity, pmho/m
23	—	2.4
35	88	3.4
45	90	4.3
55	95	4.9
70	103	6.2
80	108	7.6

*Texaco MIL-H-5606, 38°C.

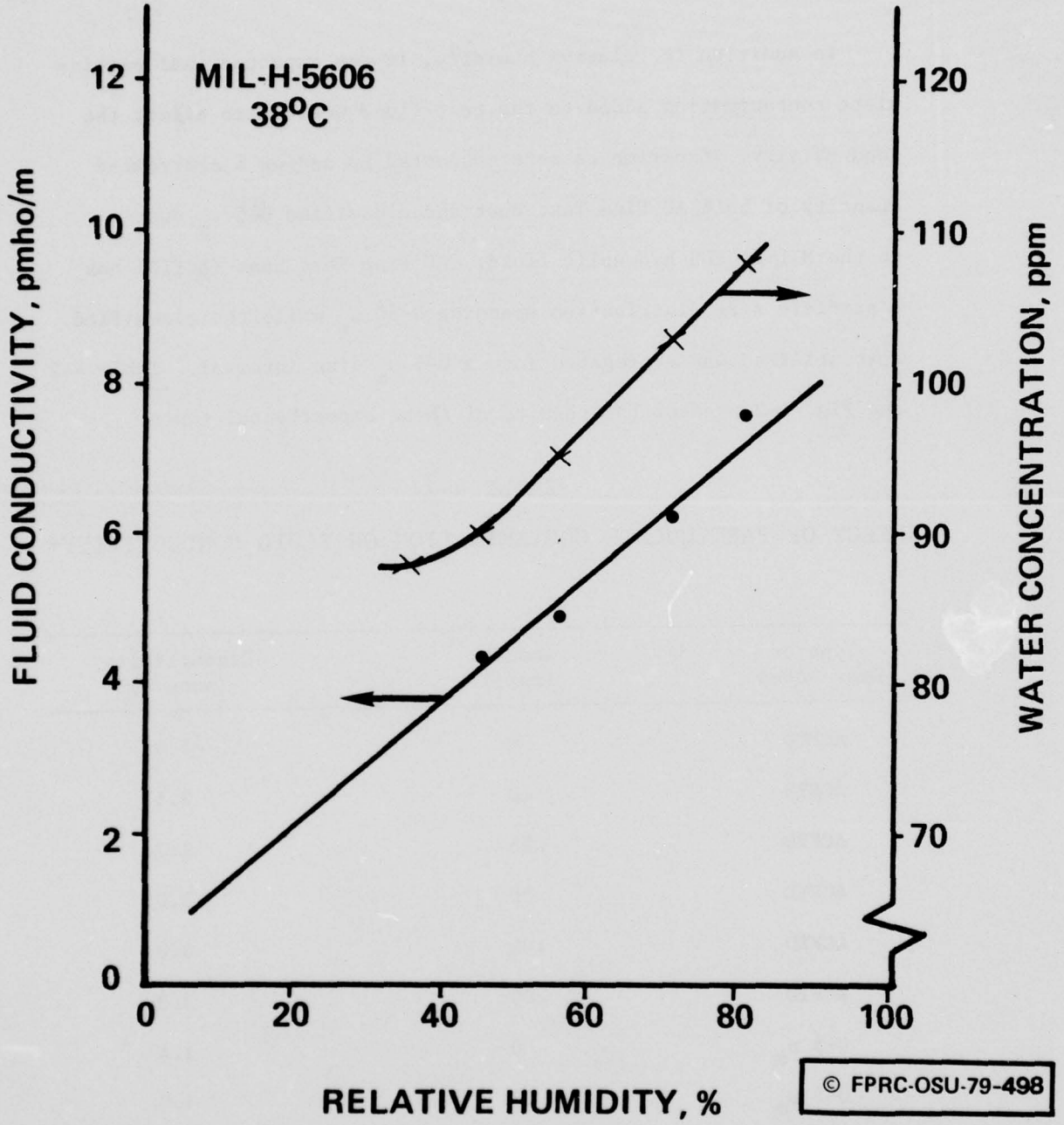


Fig. 4-1. Effect of Relative Humidity on Conductivity and Water Content.

In addition to relative humidity, it was expected that particulate contamination added to the test fluid would also affect the conductivity. Experiments were conducted by adding a controlled quantity of both AC Fine Test Dust and classified 0-5 μ_m dust to the MIL-H-5606 hydraulic fluid. AC Fine Test Dust (ACFTD) has a particle size distribution spanning 0-80 μ_m while the classified dust utilized was segregated into a 0-5 μ_m size interval. Table 4-2 and Fig. 4-2 present the results of these experimental tests.

TABLE 4-2

EFFECT OF PARTICULATE CONTAMINATION ON FLUID CONDUCTIVITY*

Type of Dust Added	Amount (mg/l)	Conductivity (pmho/m)
ACFTD	0	2.4
ACFTD	10	2.5
ACFTD	25	2.7
ACFTD	50	3.0
ACFTD	100	3.0
ACFTD	200	3.3
0-5 μ_m	0	2.4
0-5 μ_m	50	4.0
0-5 μ_m	100	5.1

* MIL-H-5606, 23% R.H., 38°C.

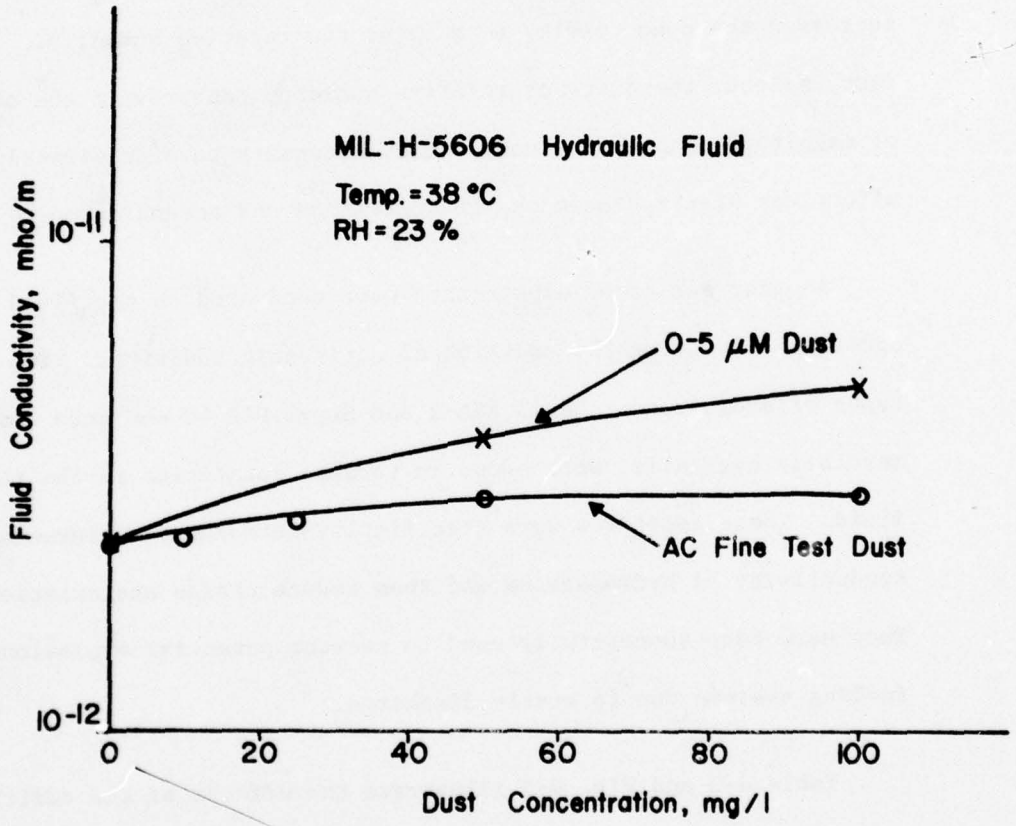


Figure 4-2. Effect of Particulate Contamination on Conductivity

It can be seen from Table 4-2 and Fig. 4-2 that dust added to the base test fluid does indeed increase the measured conductivity. The 0-5 μ_m dust increased the conductivity a greater amount than the AC Fine Test Dust of the same concentration by weight. This is probably due to the much larger total surface area presented by an equal weight of smaller particles. Neither the addition of the full distribution AC Fine Dust nor the classified 0-5 μ_m fraction increased the conductivity as much as the relative humidity. In fact, neither the dust nor relative humidity can provide the order of magnitude changes in conductivity necessary to significantly affect the electrostatic charge generation and accumulation.

Another series of experiments were conducted on the fluid conductivity involving the addition of antistatic additives. Two types of additives — Shell ASA-3 and Ethyl DCA 48 — both commercially available, were added in various quantities to the test fluid. These additives were specifically formulated to increase the conductivity of hydrocarbons and thus reduce charge accumulation. They have been successfully used to prevent potential explosions in fueling systems due to static discharge.

Table 4-3 and Fig. 4-3 illustrate the effects of the additives on the equilibrium conductivities of the hydraulic fluid. Both additives have similar effects which are much more dramatic than either relative humidity or particulate contamination. By adding a small quantity of either of the additives, the conductivity can be

increased by orders of magnitude. On a relative basis, humidity changes or particulate impurities can only provide a small conductivity improvement.

TABLE 4-3

EFFECT OF FLUID ADDITIVES ON CONDUCTIVITY*

Additive	PPM Additive	Temperature	Conductivity pmho/m
ASA-3	2	38°C	16.9
ASA-3	5	38°C	39.9
ASA-3	10	38°C	65.4
ASA-3	40	38°C	247
DCA-48	0.05	21°C	4.8
DCA-48	0.1	21°C	5.3
DCA-48	0.5	21°C	5.8
DCA-48	1.0	21°C	7.0
DCA-48	5	21°C	22.6
DCA-48	10	21°C	33.5
DCA-48	50	21°C	89.3
DCA-48	100	21°C	136
DCA-48	500	21°C	372

* Texaco MIL-H-5606, 35% R.H.

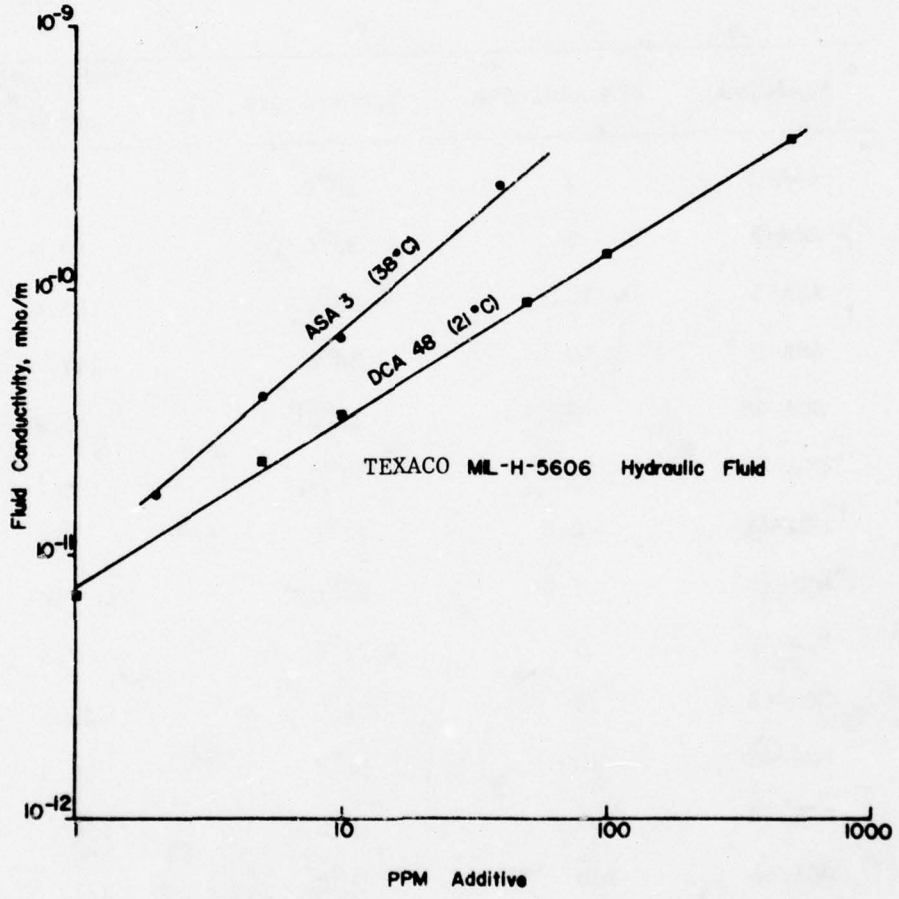


Figure 4-3. Effect of Additive Concentration on Fluid Conductivity

Because MIL-H-5606 is not the only fluid utilized in multi-pass filter test systems and actual hydraulic systems, a series of tests were conducted to determine the base conductivity of other types of oils as well as the effects of additives upon their conductivity. Table 4-4 presents the results of the conductivity measurements on the various hydraulic fluids. It can be seen that the two oils (Gulf and J.I. Case) which are normally used in mobile hydraulic equipment have much higher conductivities than the aircraft MIL-H-5606 oils. This is probably due to the presence of more additives in the fluids. The conductivity of the different brands of MIL-H-5606 oil even varied by almost an order of magnitude.

TABLE 4-4

CONDUCTIVITIES OF VARIOUS HYDRAULIC FLUIDS

Fluid Manufactures	Fluid Type	Conductivity, pmho/m
Texaco	MIL-H-5606	4.3
Mobil	MIL-H-5606	17.8
Exxon	MIL-H-5606	2.2
Texaco	MIL-H-5606 (USED)	16.3
Gulf	XHD, 10 wt; MIL-L-2104	13500
J.I. Case	TCH-45	23000

Figure 4-4 illustrates the effect of Shell ASA-3 additive upon the conductivity of two of the oils other than Texaco MIL-H-5606. It can be seen that the conductivities again increased with additive concentration.

Because the addition of conductivity additives appears to be the most convenient method to control the fluid conductivity and the resultant charge, two additional tests were performed to determine the stability characteristics of the additive. A test was conducted in which samples of fluid were extracted from a controlled and finely filtered test system over an extended period of time. The system filter exhibited an average 10 μm filtration ratio in excess of 500. The samples, extracted over a 1000 hour operating period, exhibited less than a 3% change in fluid conductivity without the addition of more additive. This test illustrates that the ASA-3 additive has long term stability in a recirculating filtered system.

The final test conducted involved the filtration of an ASA-3 doped fluid through a 0.22 μm membrane filter to determine whether or not the additive could be filtered from the fluid. These tests showed a slight (less than 10%) reduction in conductivity after multiple filtration through the micro-porous membrane; however, these differences were negligible with respect to the conductivity changes obtained with the insertion of additive.

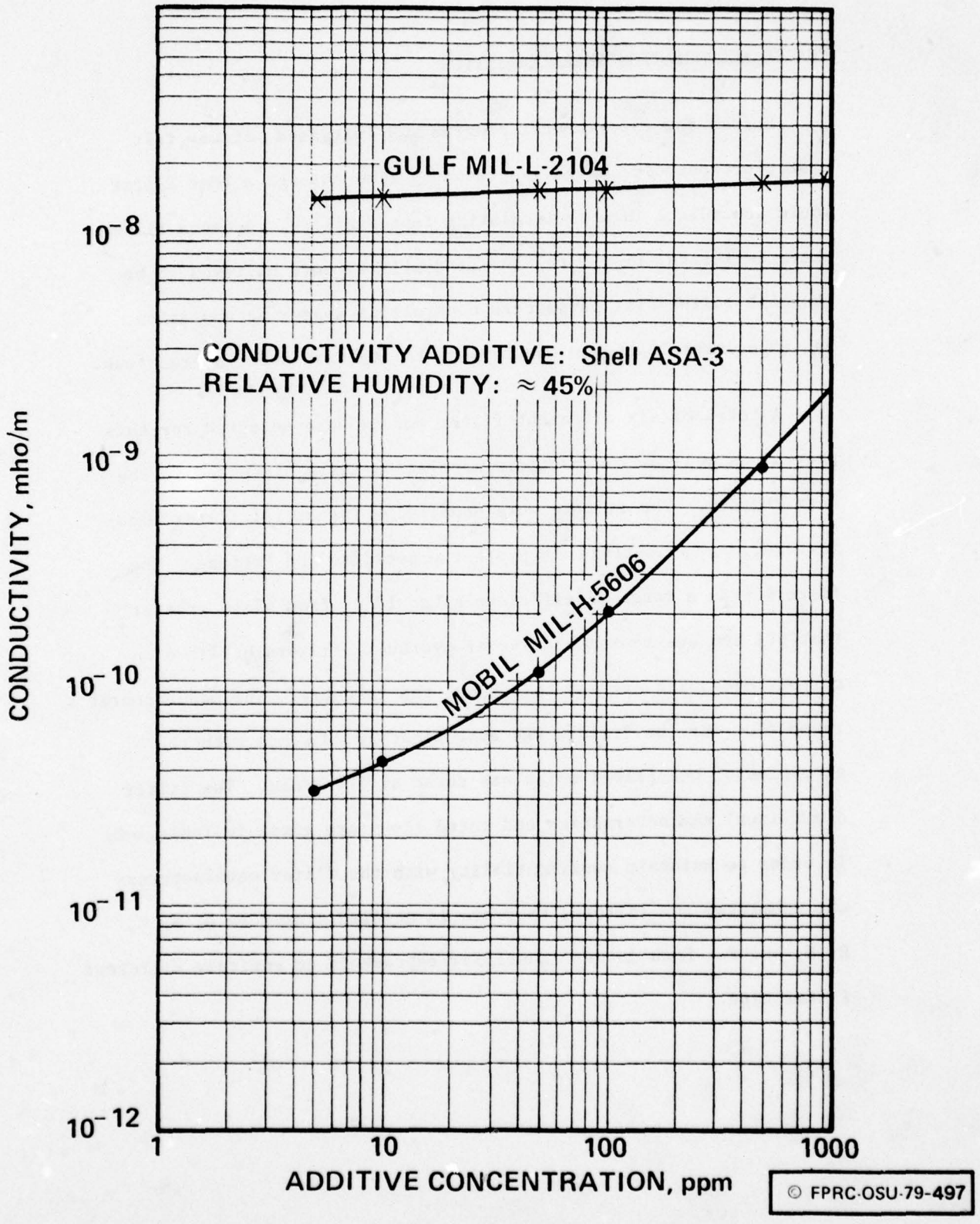


Fig. 4-4. Conductivity Additive Effect on Various Oils.

CHARGE ACCUMULATION CHARACTERISTICS

Before the experimental program was initiated, it was felt that electrostatic charge generated by flow through a test filter would accumulate in a recirculating test facility. Because filter characteristics, flow rate, and conductivity were believed to be the most influential parameters controlling charge accumulation, an experimental program was devised to evaluate such interactions.

A total of six different filter models were selected for this program as well as the particle separation tests discussed in the next chapter. The filters were donated by industrial filter manufacturers co-sponsoring this project as noted in the Preface. The filters span a range of efficiencies at 10 μ_m from 6% to greater than 98% and are representative of hydraulic filters utilized in current hydraulic systems throughout the industry. The manufacturer's rated flow for the filters was approximately 76 l/min with the exception of one filter which was rated at 114 l/min. The filter dimensional characteristics and rated flows are given in Table 4-5. In order to maintain confidentiality with the filter manufacturers who submitted the filters, they are identified simply as A, B, C, D, E, and F. Each letter identifier represents an entirely different filter type.

TABLE 4-5

TEST FILTER DESCRIPTIONS

Filter Identification	Element Length, cm	Element Width, cm	Approximate Medium Surface Area m^2	Rated Flow, l/min
A	22.8	11.2	0.768	114
B	20.2	11.0	0.728	76
C	20.2	11.0	0.628	76
D	20.2	11.0	0.465	76
E	22.9	9.8	0.850	76
F	11.0	7.8	0.243	76

CHAPTER V

EXPERIMENTAL EVALUATION OF CHARGE INFLUENCES

A large number of experimental tests were performed as a part of this investigation to establish the influence of electrostatic charge upon filter performance. A total of six different filter models were selected to cover a broad range of filtration characteristics or "effective pore size distributions." They are all representative of filters currently being specified and used in the Fluid Power Industry. The experimental program was primarily designed to evaluate the changes in separation performance and contaminant capacity which occurs when electrostatic charges are present.

FILTER TEST METHOD AND TYPICAL RESULTS

The multi-pass filter test method [27] which was originally developed at Oklahoma State University [24, 25] is recognized throughout the world as a standard method for evaluating the performance characteristics of hydraulic filters. It has received adoption by both national and international standards bodies. The test method, due to its recirculatory nature, provides a realistic representation of actual operating environments for field systems. Because of the applicability of the test results and the wide acceptance of the method, the multi-pass method was used for all evaluations in this study. Electrostatic charge density measurements were taken

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upstream and downstream of the test filter before and during each test.

The multi-pass test method requires the continuous introduction of AC Fine Test Dust into the reservoir of the recirculating filter test system. Appendices A and B describe the test facility and test method in detail. The rate of contaminant injection is controlled to result in a theoretical base upstream gravimetric level of 10 mg/l calculated by the following relationship:

Injection Rate (mg/min) = (10 mg/l) x (filter flow rate l/min)

Fluid samples are taken from the upstream and downstream lines for particulate analyses and determination of filtration ratios or separation efficiency. The test time required for the filter to plug to a designated pressure differential is recorded and the apparent contaminant capacity (ACFTD, capacity) is calculated by multiplying this time by the injection rate.

Table 5-1 represents a typical summary of data resulting from a multi-pass test. The large magnitude of information resulting from such a test (as well as the effort required to conduct the procedure) can be appreciated from Table 5-1. Particle counts were performed during this investigation at particle sizes of 2, 3, 5, 7, 10, 15, 20, 25, 30, 35, 40, and 50 micrometres on a cumulative basis. An average of three separate counts is reported. The values for beta listed in Table 5-1 are simply the ratio of the upstream count at some particle

Table 5-1.

SUMMARY OF RESULTS FOR FILTER F-7

FILTER ID: F 7 FPRC NO: 545H CHARGE: 2.67E-03 C/M**3 NO. OF SAMPLES 4

FLOW RATE = 76 L/MIN PPM = 0 RH = 37% BP = 18.8 CM. H2O

PRESSURE DIFF. (BAR): TERMINAL = 2.67 HOUSING = 0.44 CLEAN ASSEMBLY = 0.46

	2.5%	5%	10%	20%	40%	80%	100%
ASSEMBLY PRESSURES (BAR)	0.54	0.60	0.73	0.99	1.53	2.59	3.13
TEST TIMES(MIN.)	3.90	4.80	5.85	6.80	7.55	8.20	8.40

INITIAL INJECTION: .268 L/MIN. 2604 MG/L FINAL GRAVIMETRIC: 22.0 MG/L

FINAL INJECTION: .268 L/MIN. 2672 MG/L ACFTB CAPACITY: 5.9 GRAMS

TIME		>2	>3	>5	>7	>10	>15	>20	>25	>30	>35	>40	>50
5.85	UP AVG	15747.0	9674.0	5705.00	2985.10	1453.10	500.90	228.70	124.90	73.90	46.15	29.98	15.64
	DOWN AVG	3900.0	1030.0	238.60	46.80	11.80	3.14	0.77	0.48	0.48	0.31	0.15	0.00
	BETA	4.04	9.39	23.91	63.78	123.14	159.52	297.01	371.52	194.47	148.87	199.87	*****
	EFFIC.	75.23	89.35	95.82	98.43	99.19	99.37	99.66	99.63	99.49	99.33	99.50	100.00
6.80	UP AVG	16093.0	9716.0	5649.00	2939.70	1422.60	488.60	223.20	114.20	68.60	42.40	27.45	13.34
	DOWN AVG	4093.0	1065.0	254.30	53.90	17.20	5.21	1.92	0.99	0.69	0.69	0.54	0.23
	BETA	3.93	9.12	22.21	54.54	83.00	93.78	117.03	124.67	100.87	61.45	50.83	58.00
	EFFIC.	74.57	89.04	95.50	98.17	98.80	98.93	99.15	99.20	99.01	98.37	98.03	98.28
7.55	UP AVG	16600.0	10150.0	5951.00	3086.30	1517.00	523.80	236.00	120.90	69.80	40.71	25.53	12.50
	DOWN AVG	3897.0	1074.0	282.00	70.80	23.50	7.13	2.68	1.15	0.61	0.38	0.23	0.08
	BETA	4.26	9.45	21.10	43.59	64.55	73.46	88.06	105.13	114.43	107.13	111.00	156.25
	EFFIC.	76.52	89.42	95.26	97.71	98.45	98.64	98.86	99.05	99.13	99.07	99.10	99.36
8.20	UP AVG	16475.0	10251.0	6079.00	3195.60	1565.70	551.20	252.90	130.40	77.50	46.92	30.28	15.87
	DOWN AVG	3496.0	1026.0	298.90	80.80	27.50	7.36	1.99	0.61	0.31	0.23	0.15	0.00
	BETA	4.71	9.99	20.34	39.55	56.93	74.89	127.09	213.77	250.00	204.00	201.87	*****
	EFFIC.	78.79	89.99	95.08	97.47	98.24	98.66	99.21	99.53	99.60	99.51	99.50	100.00
AVERAGE BETA		4.22	9.48	21.81	48.64	75.15	91.12	128.46	154.47	143.91	107.44	103.52	169.19
AVG EFFICIENCY		76.28	89.45	95.42	97.94	98.67	98.90	99.22	99.35	99.31	99.07	99.03	99.41
10/90 AVG BETA		4.22	9.48	21.81	48.64	75.15	91.12	128.46	154.47	143.91	107.44	103.52	169.19
10/80 AVG EFF		76.28	89.45	95.42	97.94	98.67	98.90	99.22	99.35	99.31	99.07	99.03	99.41

size to the respective downstream count. The efficiency values are calculated as 100 $(1-\overline{N}_d/\overline{N}_u)$. The average values reported at the bottom of Table 5-1 are divided into two groups — the average of all samples taken during the test, and the average of only the samples corresponding to 10, 20, 40, and 80% pressure drop increases in accordance with the standard method. Additional samples were sometimes collected during the tests to provide further information about the filtration characteristics. The averages for separation efficiencies are strictly arithmetic averages, however, the average betas, because of the non-linear characteristic of the filtration ratio were calculated by $\beta_{avg} = 100/(100 - \eta_{avg})$ per the discussions in [29].

The upstream and downstream particle counts for the filter F-7 from Table 5-1 are plotted on log-log^2 coordinates in Fig. 5-1. The "spread" between the two curves is representative of the filtration ratio or efficiency and it can be seen that the distance between the lines generally increases with increasing particle sizes. For large particle sizes there is usually more random error associated with the particle counts from a statistical basis [30] because of the low particle population. Thus, the filtration ratios are large sizes (especially with high efficiencies producing a very small particle population) shows a greater degree of scatter. In addition, if any fluid leakage occurs across the element seals (from upstream to downstream), the filtration ratios show a tendency

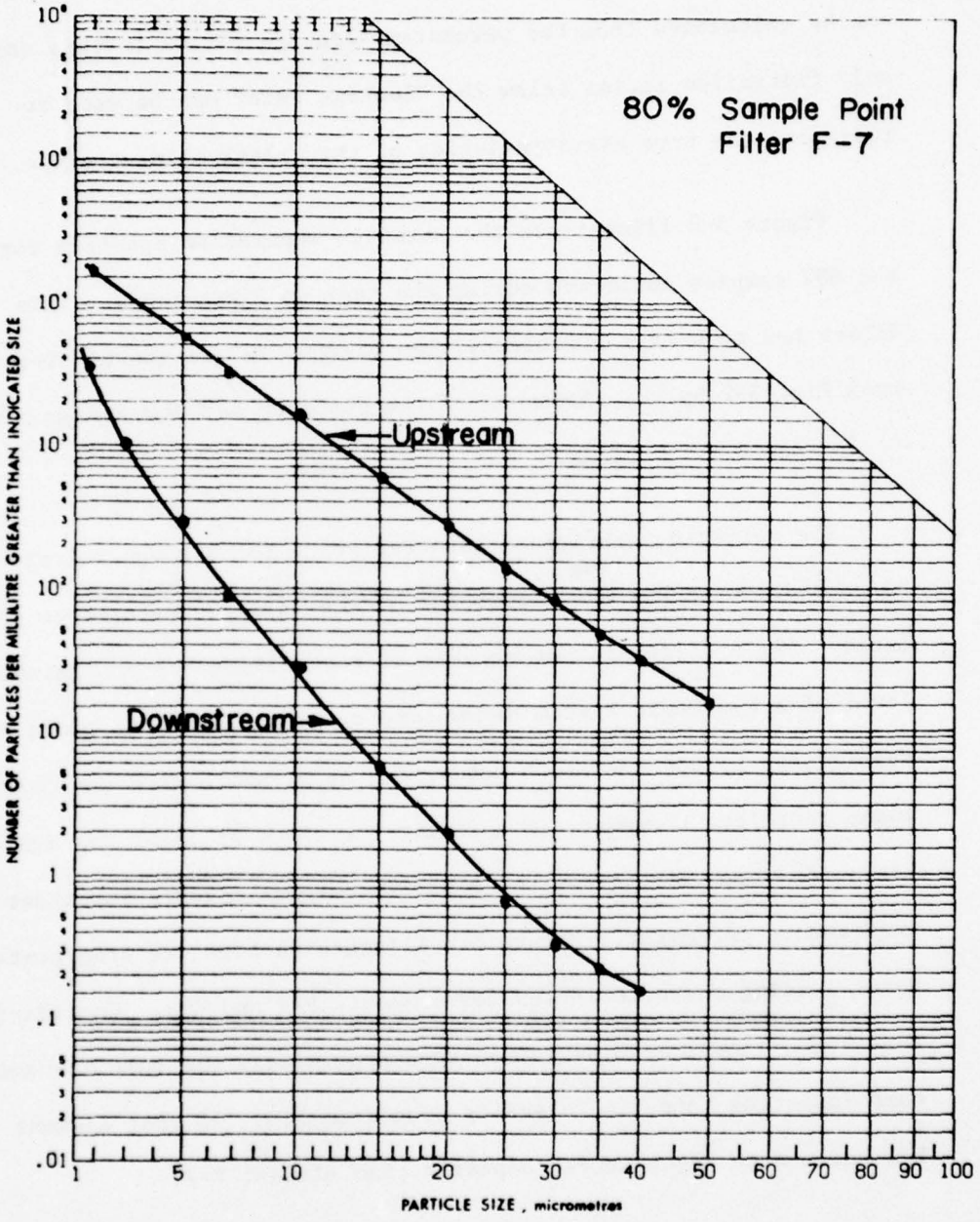


Figure 5-1. Particle Separation Spectrum for Filter F-7

to become constant for all larger particle sizes at a value which can be calculated from the percentage leakage [31]. In this case, only filtration ratios below the "leakage rato" can be used to represent the true characteristics of the filter element alone.

Figure 5-2 illustrates the particle separation spectrum for the 80% samples collected during the test on Element E-2. This filter has much less efficiency than element F-7 as can be visualized from Fig. 5-2 by the closeness of the upstream and downstream particle size distributions.

The particle separation characteristics for a given filter must be considered as the most important of the various performance parameters. If a filter is not capable of controlling the contamination level of a hydraulic system below some critical level, it would be meaningless to consider such parameters as pressure drop and contaminant capacity. However, if a filter possesses at least the desired retention characteristics, the capacity becomes a quite important performance parameter. Figure 5-3 illustrates pressure drop/contaminant loading characteristics for elements F-7 and E-2. Considering that both filters were subjected to approximately the same contaminant injection rate, it is easily seen from Fig. 5-3 that element E-2 had a much higher ACFTD capacity than element F-7.

A complete summary of the average filtration performance capabilities of the thirty-nine filters evaluated during this investi-

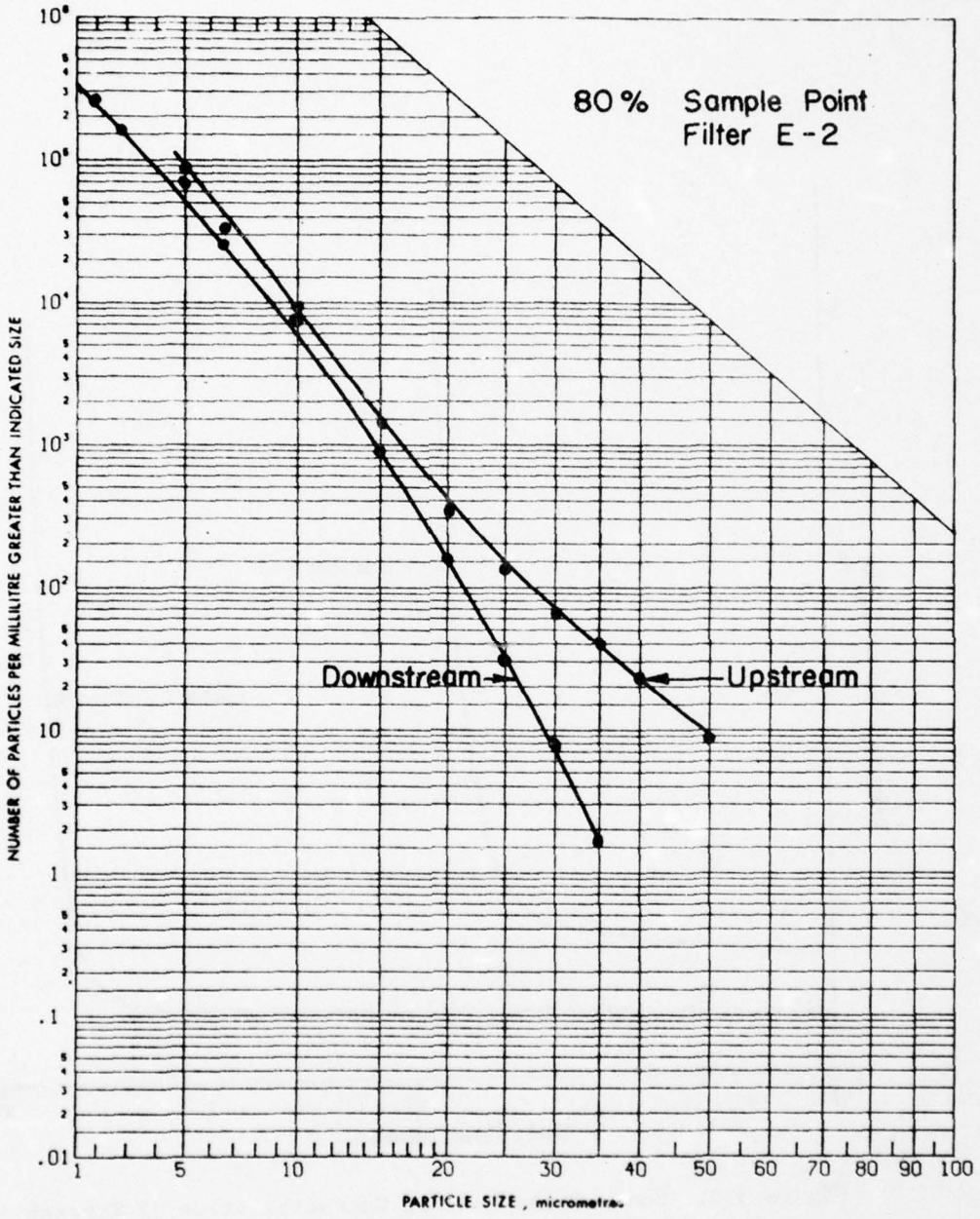


Figure 5-2. Particle Separation Spectrum for Filter E-2

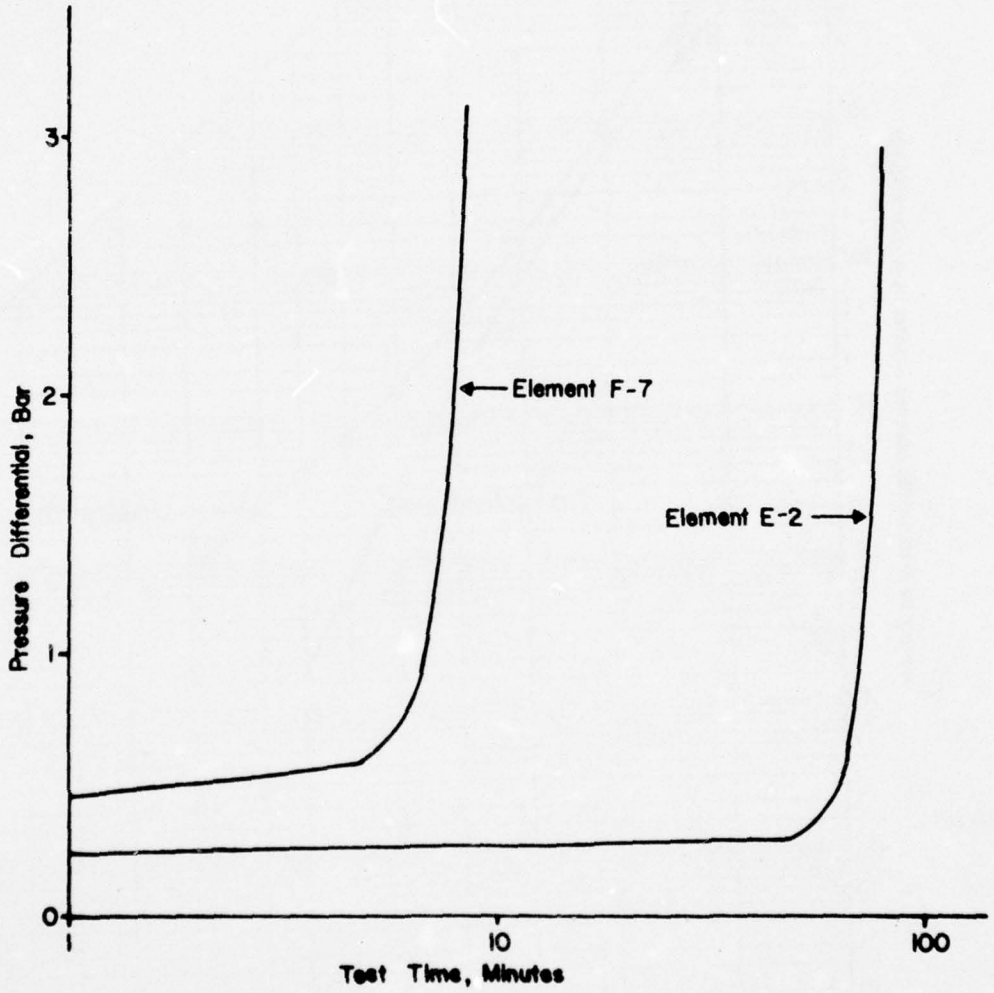


Figure 5-3. Contaminant Loading Characteristics of Typical Filters

gation is included in Appendix D. The filtration ratios, as well as efficiencies, are averages of the samples extracted at the 10, 20, 40, and 80% pressure increase points. These points were selected for the summary because the standard multi-pass procedure calls for sampling at these respective points and because these samples were available for all the filters evaluated. Each set of average filtration ratios or efficiencies thus represents a compilation of eight samples and a total of thirty-six particle counts per sample (12 particle sizes x 3 counts for each size) or a total of 288 numbers. Because of the magnitude of numbers and calculations required, all the data was entered into a computer bank for summarizing and data reduction purposes.

EXPERIMENTAL RESULTS OF CHARGE INFLUENCE TESTS

As mentioned before, a total of thirty-nine complete multi-pass tests were conducted under various charging levels. A complete summary including the measured charge density values is contained in Appendix D. There was no convenient method for controlling the electrostatic charge density at a desired value for any given test as the charge level is a function of filter properties, fluid properties, and flow rate. Tests were therefore conducted with various amounts of conductivity additive (ASA-3) in the test fluid Texaco (MIL-H-5606) in order to establish various charging levels.

During the tests, the charge densities in the fluid upstream of the test filter were measured but were always at least an order of magnitude less than the downstream fluid. Generally, the difference corresponded to several orders of magnitude as predicted by the generalized relaxation law presented in Chapter III. The downstream charge density is therefore the only value reported in the summary of Appendix D.

It was suggested to the author that perhaps the addition of the ASA-3 anti-static additive to the test fluid could increase the particle dispersancy characteristics of the fluid thus changing the apparent performance of the test filter. Past experience has shown that with the turbulent mixing (flow passing through the pump, tubing, elbows, difusers, etc.) that exists in a multi-pass test facility, AC Fine Test Dust becomes fully dispersed. A test was conducted however to determine the effects, if any, of ASA-3 upon the particle size distribution of ACFTD in MIL-H-5606 hydraulic fluid. A quantity of dust was mixed in a small recirculating test stand with new MIL-H-5606 fluid. Samples were then extracted for particle count analysis. A quantity of ASA-3 additive (40 PPM by volume) was then added to fluid and the circulation was allowed to continue for 30 minutes. Samples were again extracted for analysis. Table 5-2 presents the average results of particle counts conducted on two samples taken before and after the addition of the ASA-3. It can be seen that the particle size distribution with the ASA-3

is essentially equal to the distribution with no additives.

TABLE 5-2

DISPERSANCY CHARACTERISTICS OF SHELL ASA-3 ADDITIVE

Particle Size (μ_m)	Cumulative Particle Count/ml	
	No additive	40 PPM ASA-3
2	6398	6384
3	4319	4421
5	2636	2602
7	1373	1401
10	646	670
15	206	208
20	85.0	91
25	40.3	41.5
30	21.5	20.3
35	11.3	12.7
40	6.8	6.9
45	1.7	2.5

It can be noted from the data summary in Appendix D that several filters of each model were evaluated with extremely low downstream charge densities. For instance, with filter A, the maximum charge density obtained was with no conductivity additive and was equal to

11.7×10^{-3} C/m^3. The minimum level was 1.87×10^{-7} C/m^3 or almost five others of magnitude lower. The extremely low charge measurements were difficult to measure accurately and variations of up to 100% were not uncommon. However, when the charge level was approximately 10^{-4} C/m^3 or greater the charge measurements were highly repeatable and successive reading would usually fall within 5% of the average value. The primary reason for the large number of data points at the low charge values was the fact that the effect of the conductivity additive on a particular filter was unknown prior to the addition of the ASA-3; thus, it was difficult to determine the required concentration of additive in advance. Various levels of additive from 2 PPM up to 40 PPM were used during the investigation and the filters were evaluated at each additive level. There was a large time delay from the time a test was conducted until particle counts were completed which also increased the difficulty in determining which levels of charge resulted in an effect on the filter performance.

In order to establish the effect of electrostatic charge upon the filter's particle separation capabilities for various particle sizes, it is more convenient to examine particle counts and efficiencies on an incremental size basis rather than cumulative. This allows the isolation of any influences on a given particle size or size range. Appendix D also includes a summary table of filter separation characteristics based upon size interval calculations. The values presented are similar to filtration ratios except on a

size interval basis and are called "retention ratios" to prevent confusion with the cumulative parameter. The retention ratios are calcualted by the following relationship:

$$\Phi_{D_1-D_2} = \frac{\overline{N}_{u_1} - \overline{N}_{u_2}}{\overline{N}_{d_1} - \overline{N}_{d_2}}$$

where

Φ = retention ratio for particle size interval $D_1 - D_2$

D_i = particle size

$\overline{N}_{u_i}$ = particles greater than size D_i in upstream fluid

$\overline{N}_{d_i}$ = particles greater than size D_i in dowstream fluid.

Figures 5-4 through 5-9 illustrate the relationship between average retention ratio and downstream charge density for the six filter models evaluated. To prevent the larger number of data points at extremely low charge levels from biasing regressions, an average value of the retention ratios for these charges were plotted at zero charge on Figs. 5-4 - 5-9. Charge values below 10^{-4} C/m^3 were considered alike and the retention ratios were averaged to produce a best estimate of the zero charge point. Typical coefficients of variation (100% x ratio of sample standard deviation to sample mean) for this averaging fell between 2% and 10% for retention ratios with a value under two. The variation increased in

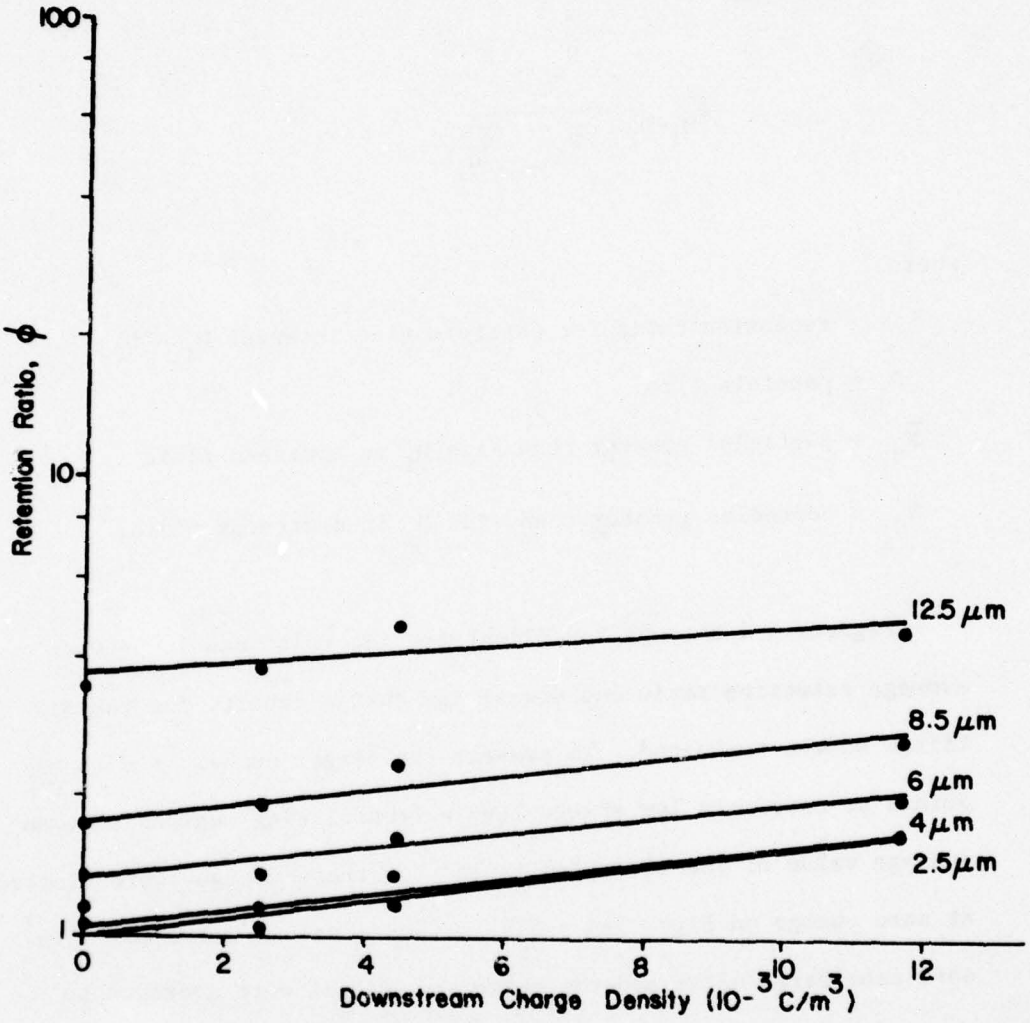


Figure 5-4. Charge Effects on the Retention Characteristics of Element A

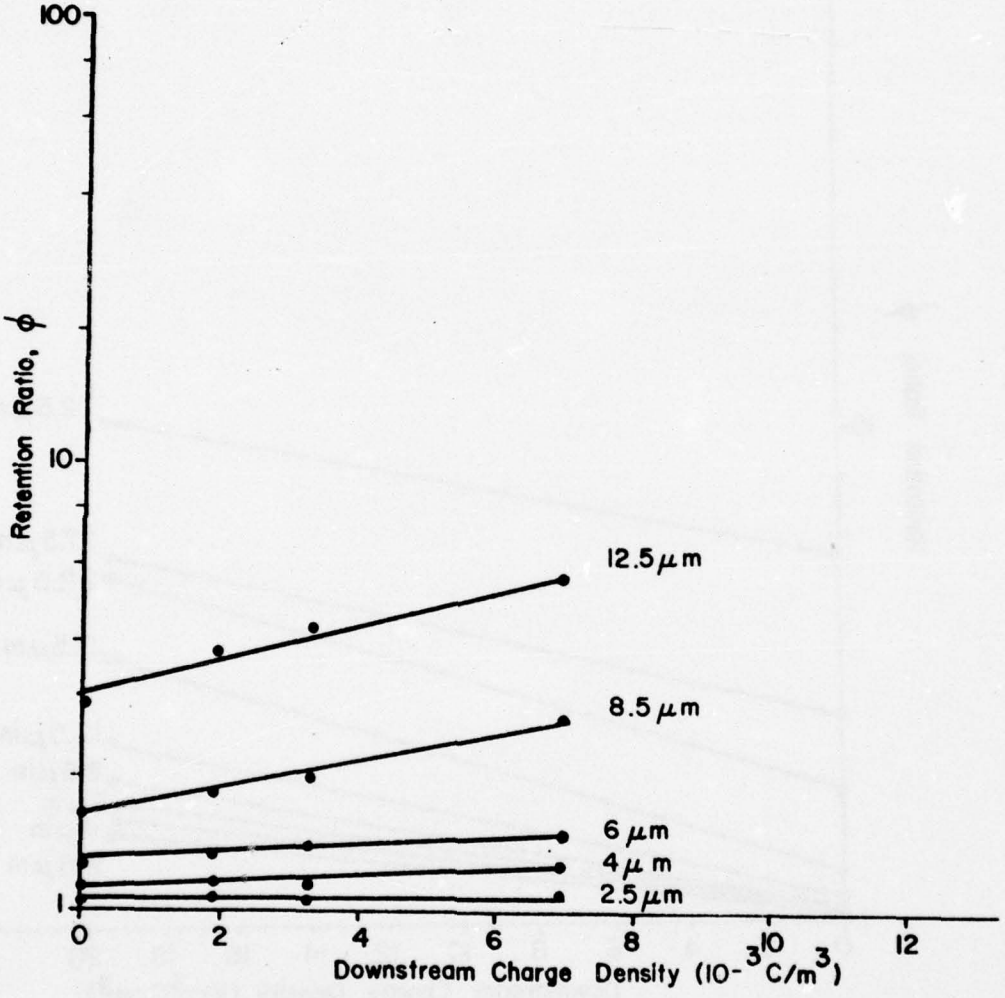


Figure 5-5. Charge Effect on the Retention Characteristics of Element B

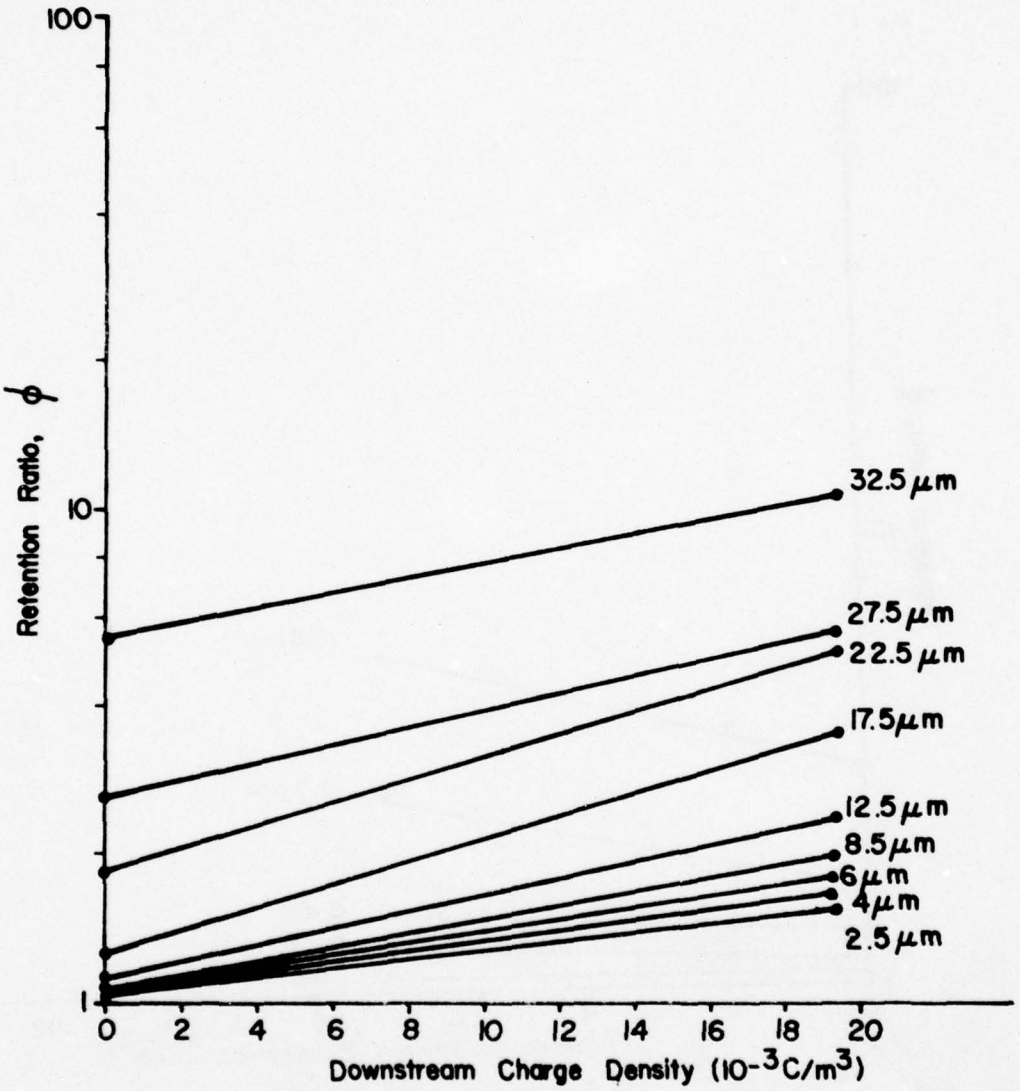


Figure 5-6. Charge Effect on the Retention Characteristics of Element C

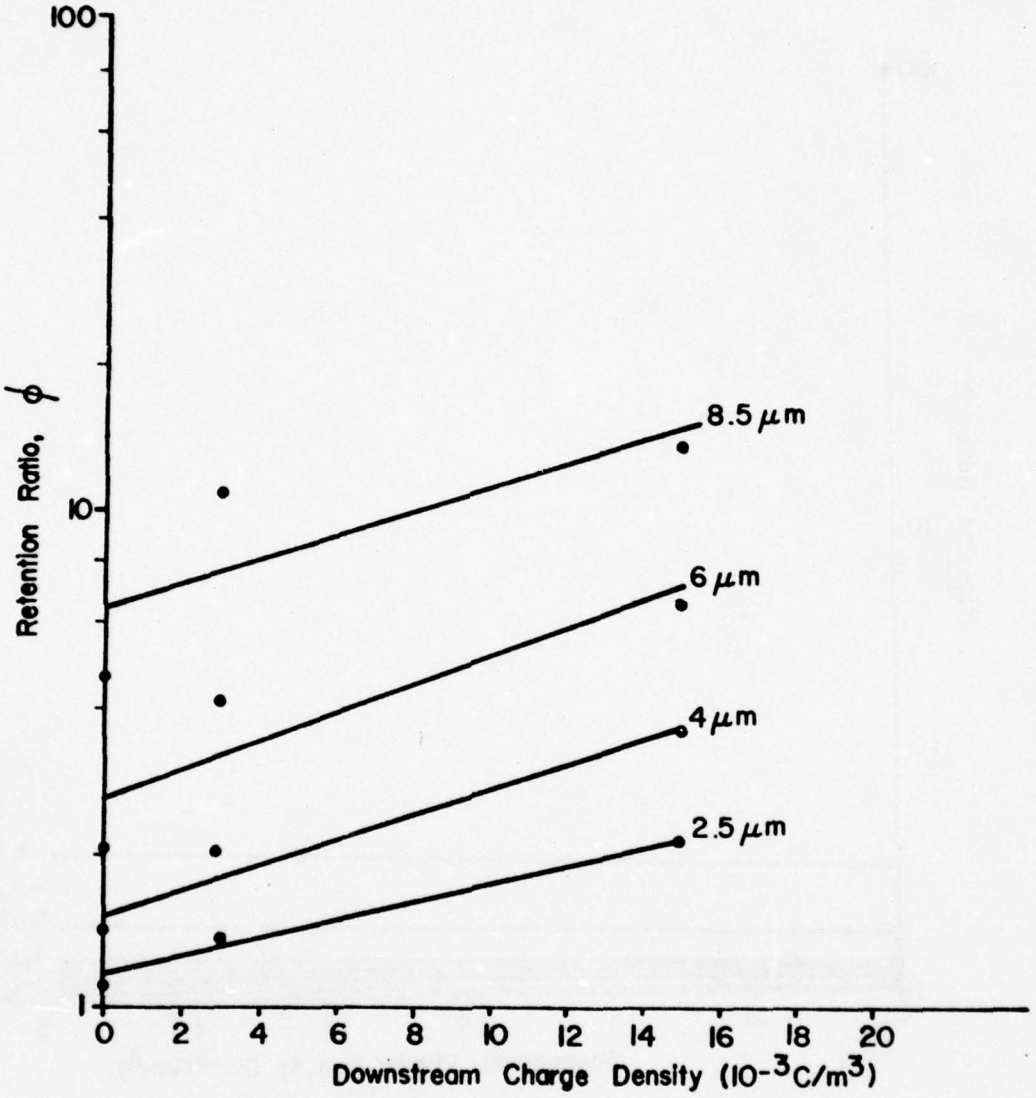


Figure 5-7. Charge Effect on the Retention Characteristics of Element D

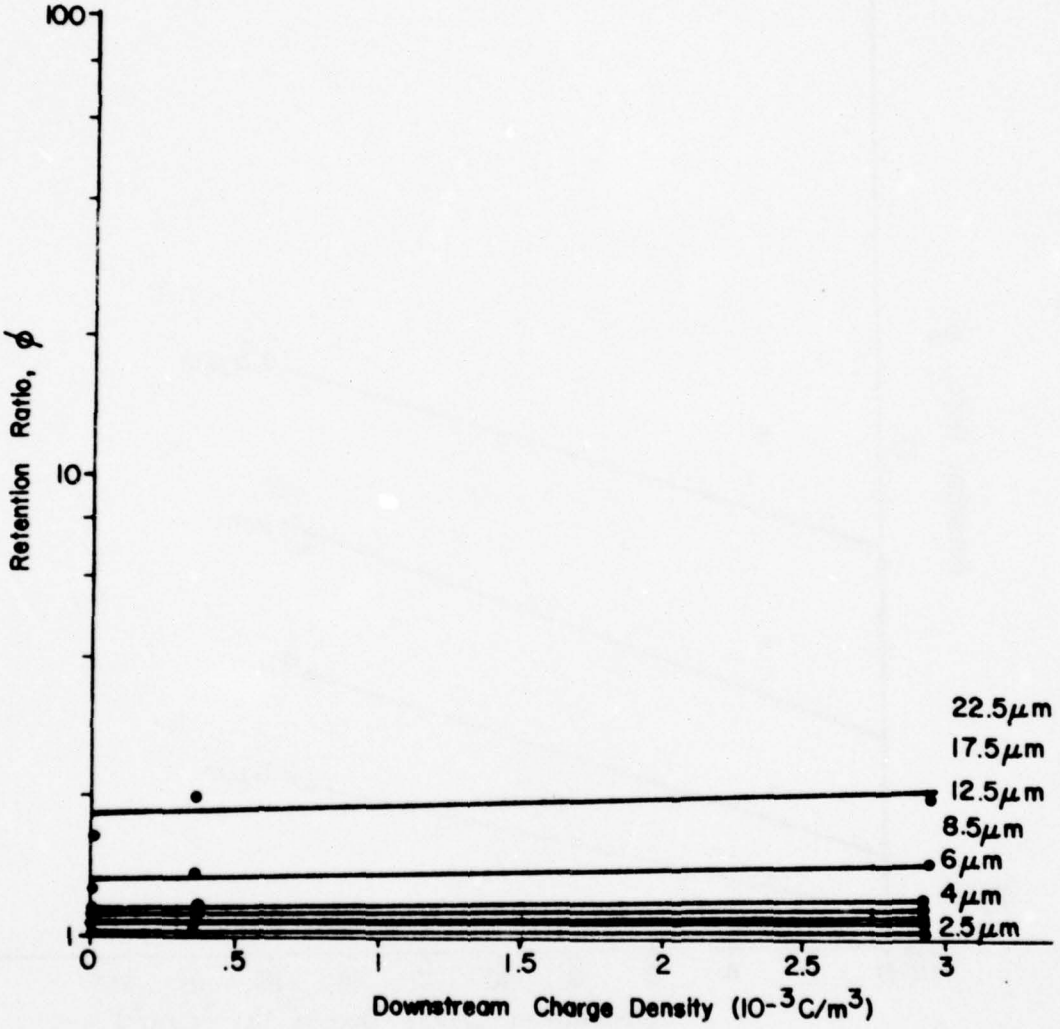


Figure 5-8. Charge Effect on the Retention Characteristics of Element E

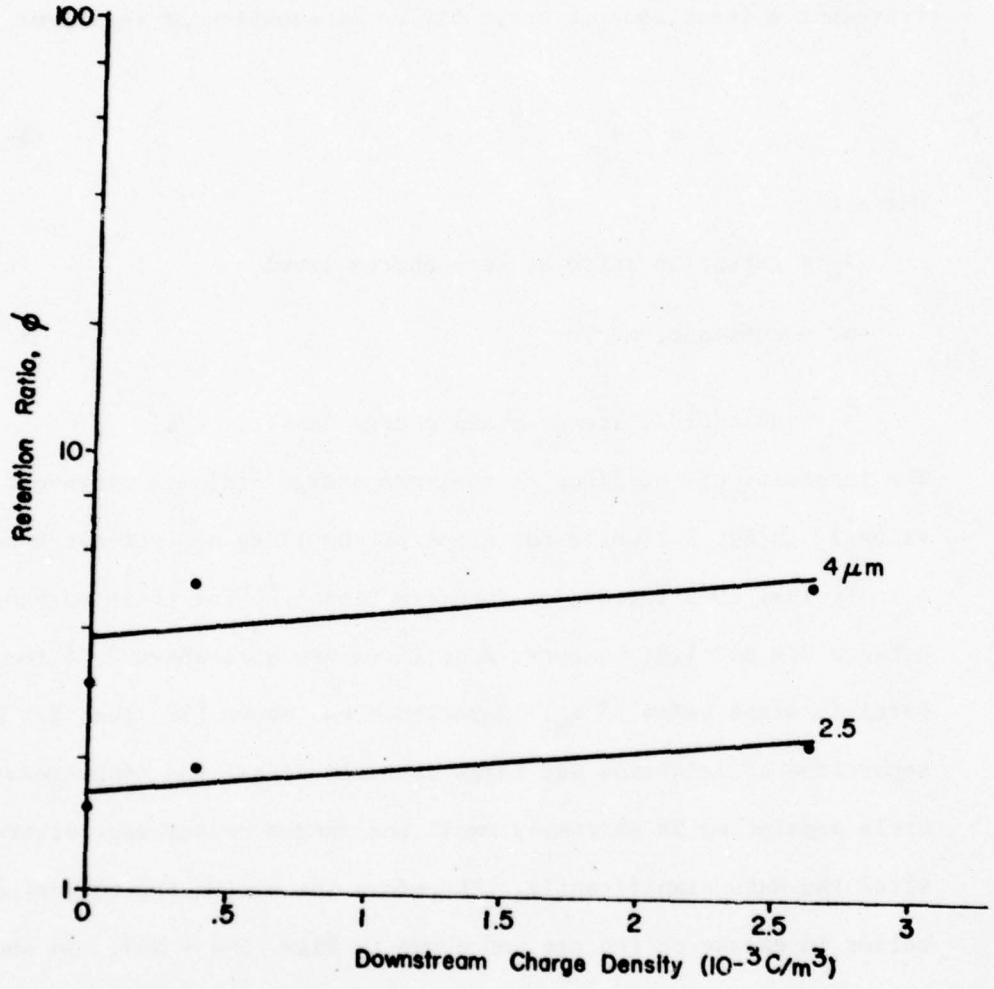


Figure 5-9. Charge Effect on the Retention Characteristics of Element F

most instances for higher retention ratios and at higher particle sizes.

The lines drawn through the data points in Figs. 5-4 - 5-9 represent a least squares curve fit to an equation of the form:

$$\Phi = \Phi_o e^{\alpha_1 \hat{s}_d} \tag{5-1}$$

where

Φ_o = retention ratio at zero charge level

α_1 = constant, m^3/C

$\hat{s}_d$ = downstream steady state charge density, C/m^3

The intercept of the lines at the zero charge ordinate represent the value Φ_o in Eq. 5-1 while the slope of the lines are reflected by α_1. Typical coefficients of determination (r^2) for these curves fell between 0.5 and 1.0; however, most r^2 values were above 0.85 for particle sizes below 15 μ_m. Experience has shown [30] that for high separation efficiencies and large particle sizes, the downstream particle population is extremely small and random measurement errors may alter the data significantly. Therefore the curves for retention ratios in excess of ten are not shown in Figs. 5-4 - 5-9. In such instances, the coefficients of determination were generally below 0.5.

An additional result from the charge influence tests was ACFTD capacities for the various filters and charge levels. Figure 5-10 illustrates the data collected from the experimental program. Again,

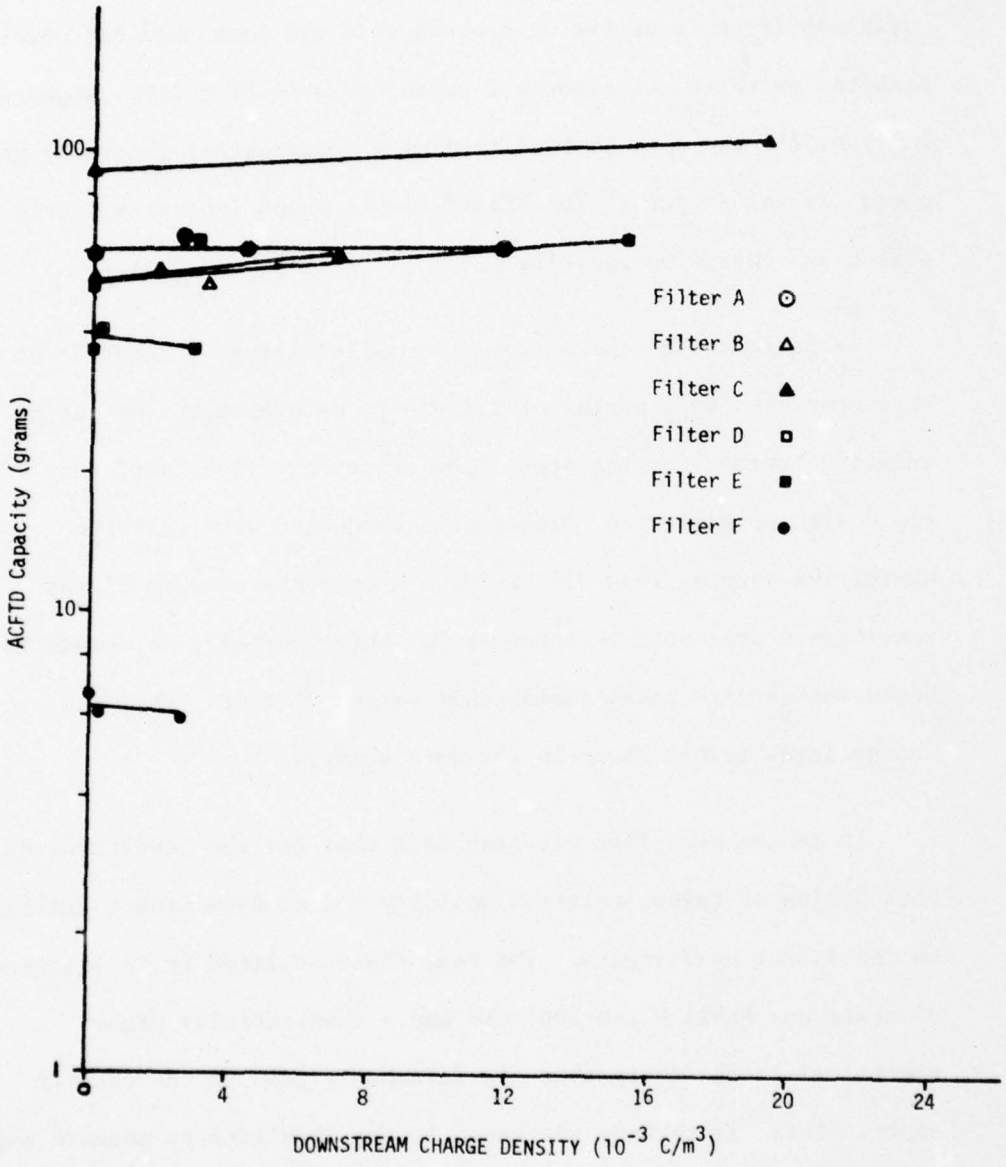


Figure 5-10. Effect of Charge Density on ACFTD Capacity

the capacity values for filters with approximately zero charge were averaged and plotted as one point at zero. The coefficients of variation for this averaging process were all less than ten percent with the exception of element F which had a 14.8% C.O.V. Regression analyses for the data plotted (semi-log coordinates) generally produced low values for r^2 and "flat" slopes which indicates little effect of charge on capacity.

In addition to the various charging level tests, a series of tests were conducted on a series of filters to determine the effect of relative humidity on the separation efficiency with "new" test fluid with no additives. Tests were conducted with relative humidities varying from 32% to 70%. A complete summary of the results are presented in Appendix D. Unfortunately, no charge level measurements were taken during this series of tests, thus the charge level is not shown in the data summary.

It can be seen from the test data that for the conditions of this series of tests, relative humidity had no significant influence on the filter performance. The test fluid utilized in this series of tests was Mobil MIL-H-5606 and had a substantially higher electrical conductivity than the Texaco oil used in the earlier tests. This is perhaps the cause of the inability to measure any change in filter separation performance with varying relative humidity.

CHAPTER VI

APPLICATIONS AND EXTENSIONS OF THE INVESTIGATION

MULTI-PASS TEST INFLUENCE

It has been suspected that the results of a "standard" multi-pass filtration performance test [27] on a hydraulic filter can be influenced by electrostatic charge effects. The results of this investigation certainly confirm these suspicions by illustrating the improvement in particle separation characteristics which can be obtained with electrostatic charge influences. Because of these effects, the multi-pass method as it is now written can not always be relied upon to give accurate and unbiased filtration performance data. Any user of the procedure could unknowing or "intentionally" influence the results by conducting tests under conditions conducive to electrostatic charge generation. Such conditions would be the use of new test fluid and operation in a low humidity environment.

The results of this study revealed that the filtration performance of hydraulic filters is significantly improved under the presence of electrostatic charge especially when charge density levels exceed 10^{-4} C/m^3 immediately downstream of the filter. In terms of the filtration ratio, which is based upon cumulative particle counts, and is most familiar to the Fluid Power Industry, the improvement can be summarized by Fig. 6-1. The values plotted are

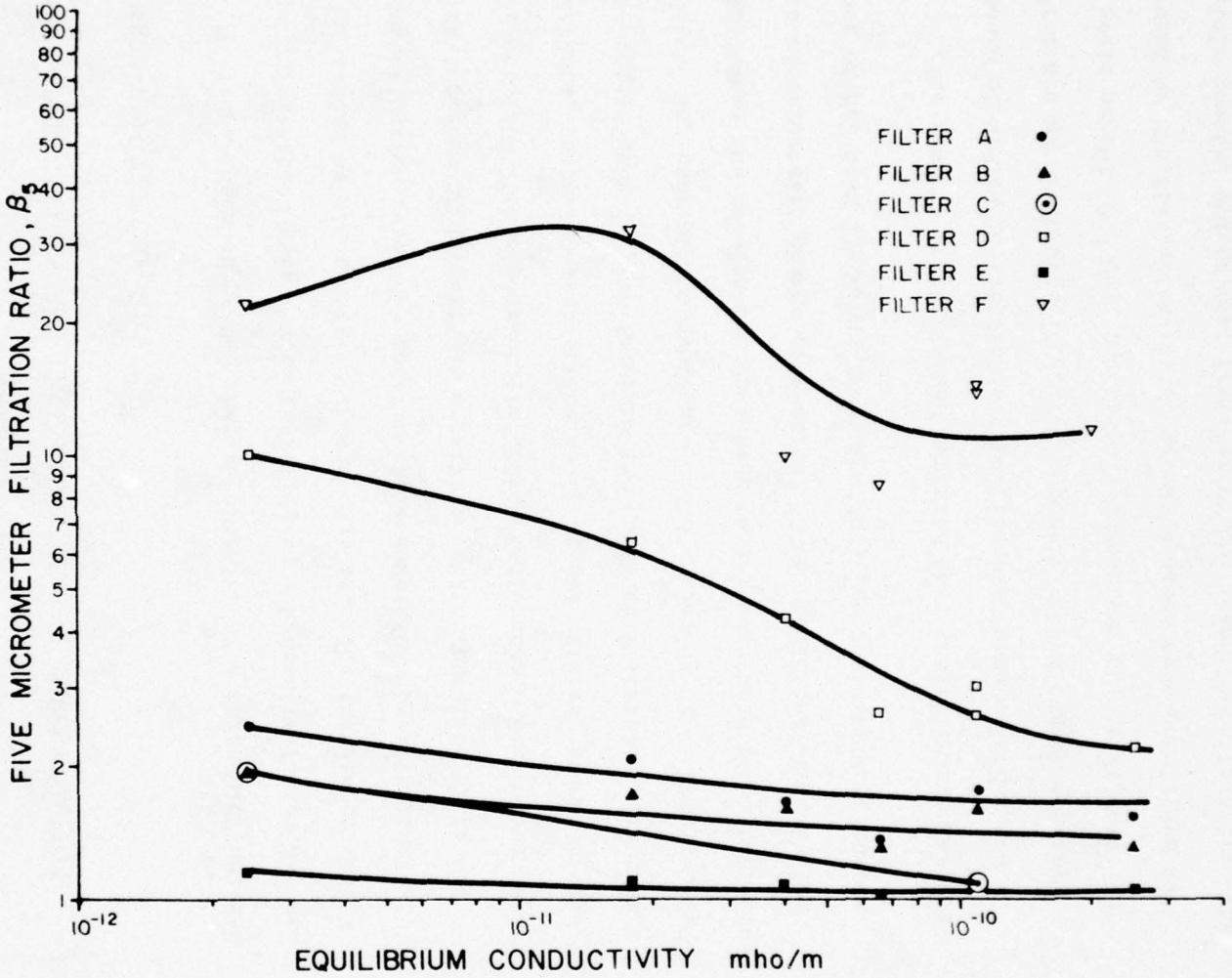


Figure 6-1. Conductivity Influence on Filtration Ratio

the average five micrometre filtration ratios for the various conductivities utilized in the experiments. It can be seen from Fig. 6-1 that as the conductivity is decreased, the filtration ratios increase. This is due to increased charge densities as shown in Fig. 6-2.

It can be concluded from the data in Fig. 6-2 that the charge levels for all filters was extremely low (less than 10^{-5} C/m^3) when the equilibrium conductivity was 10^{-10} mho/m or greater. Figure 6-1 illustrated for conductivities above approximately 10^{-10} C/m^3 that the filtration ratios are essentially constant at some minimum value. The fluid equilibrium conductivity can easily be controlled by the addition of a conductivity improver additive such as the ASA-3 or DCA-48 used in this investigation. With the ASA-3 additive, the conductivity can be increased above 10^{-10} C/m^3 for amounts in excess of approximately twenty PPM by volume.

It would be recommended from the results of this investigation that the multi-pass test method of [27] be modified to include the addition of ASA-3 additive to the test fluid by an amount in excess of 20 PPM. The stability of the additive to perform over long periods of time is unknown; therefore, it would also be recommended that the minimum amount of additive be added to the test system before each filter evaluation. The experiments conducted with various relative humidities in the surrounding environment showed that the higher humidities have a tendency to reduce charge influences; however, it is felt that humidity changes alone can not eliminate the

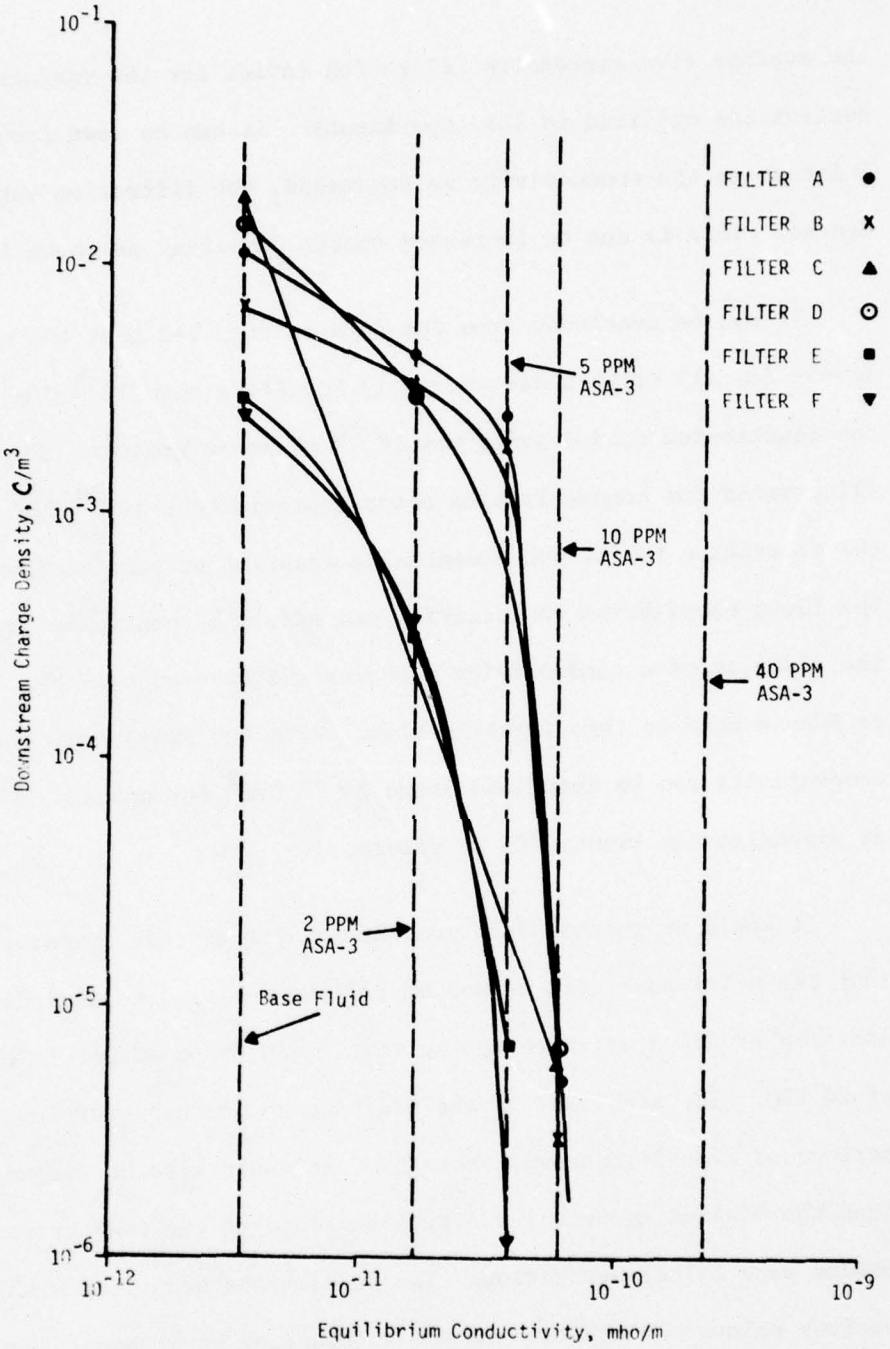


Figure 6-2. Conductivity Influence on Charge Density

electrostatic charge below a significant level. Only in the presence of conductivity additives was this accomplished.

Based upon the results of this investigation, an announcement was made to the Filter and Separator Committee of the National Fluid Power Association. This announcement stated that electrostatic charge can indeed influence filter test results and suggested that a change be made in the standard test method. The following statement was proposed for inclusion in the test procedure:

"Insure that the electrostatic charge density in the fluid immediately downstream of the filter element is less than 10^{-6} Coulombs per cubic metre. The charge density may be measured in accordance with ASTM D2679 or the low charge density may be obtained by adding sufficient anti-static additive to the test fluid. It has been determined that the addition of 40 ppm by volume of Shell ASA-3 or 500 ppm by volume of Ethyl DCA 48 will eliminate electrostatic charge influences. Other methods, if proven, may also be utilized for reducing the electrostatic charge. The fluid additives mentioned may be obtained from the following addresses: Shell ASA-3, Shell Chemical Co., 110 W. 51st Street, New York, NY 10019 and Ethyl DCA 48, Ethyl Corporation, Attn: P.C.D. Distribution Services, 451 Florida, Baton Rouge, LA 70801, 504-328-0131." It is felt that the adoption of this change to the standard multi-pass test method will produce higher repeatability, reproducibility, and industry-wide confidence levels.

RECOMMENDATIONS FOR FURTHER STUDY

This investigation has resulted in a great deal of valuable information relative to the performance of hydraulic fibrous filters with electrostatic charge influences. In order to continue the advancement of this field, the following related investigations are recommended for future study:

1) Further experimentation should be conducted to determine the degree of influence of even higher charge generations than were used in this study. The degree of improvement of filtration characteristics appears to be unlimited if charge levels are of sufficient magnitude.

2) An investigation of the dynamic retention characteristics for filters utilizing the electrostatic charge capture mechanism is an important extension of the current study. It is known that when subjected to cyclic or unsteady flow, many hydraulic filters have a tendency to retain less particulate contamination than under steady flow conditions. The ability for a filter to retain particles captured by electrostatic forces is a necessary area of investigation.

3) Finally, in order to maintain the wide acceptance and confidence that industry is gaining in the multi-pass test method, it is suggested that the recommendations made in this study be implemented as soon as possible.

CHAPTER VII

SUMMARY AND CONCLUSIONS

SUMMARY

Electrostatic charge forces have long been recognized as a major factor in aerosol filtration. It was generally thought however that such forces were not of significance in hydraulic filtration due to the much higher viscosity and drag forces created by the surrounding fluid. It was recently discovered through continued use of the standard multi-pass hydraulic filter evaluation method [27] that, under specific conditions, electrostatic charge generation was high and filter performance was changed.

The original objectives of this investigation were to examine, both theoretically and experimentally, the charge generation and accumulation characteristics of a recirculating filtered hydraulic system. In addition, the influence of this electrostatic charge upon filter performance was to be studied.

A thorough literature survey revealed a large magnitude of published information relative to the generation of electrostatic charge by fueling systems. None of these studies were directly applicable to recirculating type systems as encountered in the hydraulics field. A generalized charge relaxation and accumula-

tion law was derived and experimentally verified for a multi-pass filter test facility. Experiments were conducted to determine the various effects of fluid conductivity, fluid additives, particulate impurities, relative humidity, and flow rate.

A large filter testing program was conducted to determine the effects of electrostatic charge on these filter performance parameters. The results indicated that the separation efficiency was almost always increased with increasing charge.

CONCLUSIONS

From the investigation described in the preceding chapters, a number of conclusions can be made. The following list summarizes the major accomplishments and conclusions:

1) A generalized model for the accumulation and relaxation of electrostatic charge in a recirculating system was presented to include concepts from both the exponential and hyperbolic relaxation laws.
2) Experiments were conducted to determine the various interactions between electrostatic charge generation, fluid conductivity, surrounding environment relative humidity, particulate contaminants, and the addition of fluid conductivity improver additives. The results show that although each parameter has some influence on charge generation, the conductivity additives have a much broader predictable effect.

3) Experimental evaluations were performed to determine the influence of electrostatic charge upon the apparent contaminant capacity of hydraulic filter elements. Although slight trends towards increasing capacity with increasing charge were noted, no significant conclusions can be drawn.

4) The influence of electrostatic charge upon the filter particle separation characteristics was determined experimentally. It can definitely be concluded that electrostatic charge forces can become a dominant factor in hydraulic fluid filtration by fibrous filters. The particle separation characteristics are improved as electrostatic charge densities are increased.

5) Recommendations were made for modifying the "standard" multi-pass filter test method by including the addition of a conductivity additive to the test system fluid before performing a test. This will increase the conductivity and thus reduce the charge generation below significant levels.

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APPENDIX A

MULTI-PASS HYDRAULIC FILTER TEST FACILITY

All filter tests which were performed as a part of this research investigation were conducted utilizing the multi-pass test facility illustrated in a simplified schematic as shown in Fig. A-1. A detailed circuit schematic showing all components is presented in Fig. A-2.

The main filter test system was essentially composed of a reservoir hydraulic gear pump and circulating system with controls for temperature and flow rate. The facility was constructed of nearly all metallic components including lines and fittings. The tubing in the test circuit had a nominal size of 2.54 cm (1 inch) with an inside diameter of 2.21 cm. Thus the cross sectional or flow area of the tubing was 3.84×10^{-4} m^2. The approximate lengths of lines for the test circuit are shown in Fig. A-1.

The contaminant injection system consisted of a reservoir, pump, and appropriate controls. During testing the injection contamination level may be extremely high (several thousand mg/l concentration); therefore, the injection pump was a centrifugal type which has been found to be relatively insensitive to contaminant. The primary purpose of the injection circuit was to maintain a uniform contaminant suspension with a constant particle size distribution. This contaminant slurry is injected into the main filter test system during a complete multi-pass test.

In addition to the test and injection systems, the multi-pass facility also consisted of a contaminant clean-up system. The

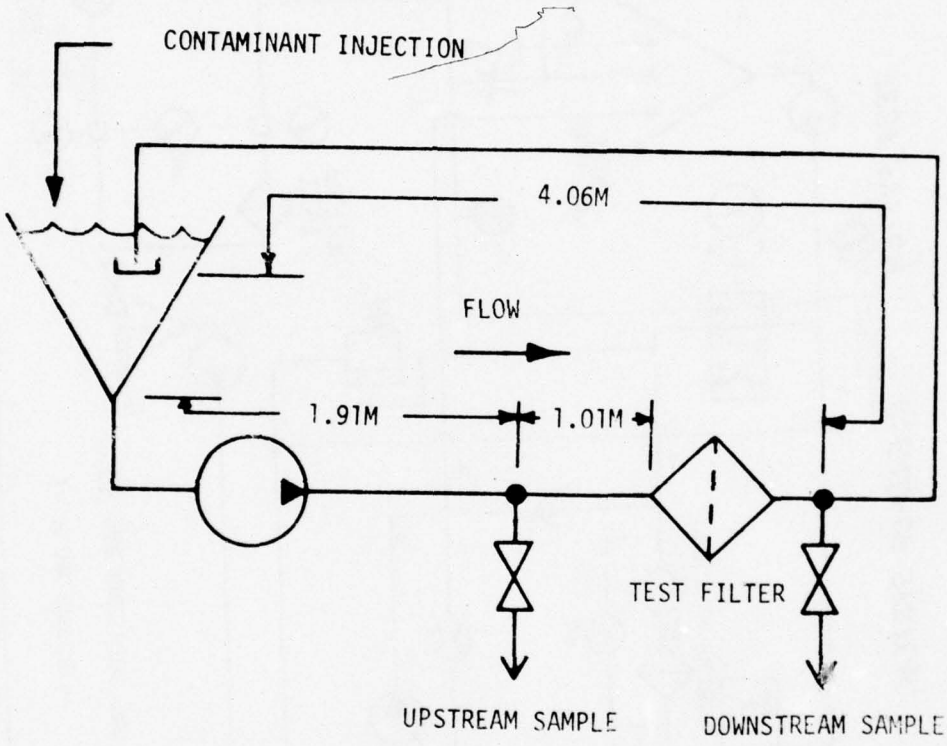


Figure A-1. Simplified Multi-Pass Circuit Schematic

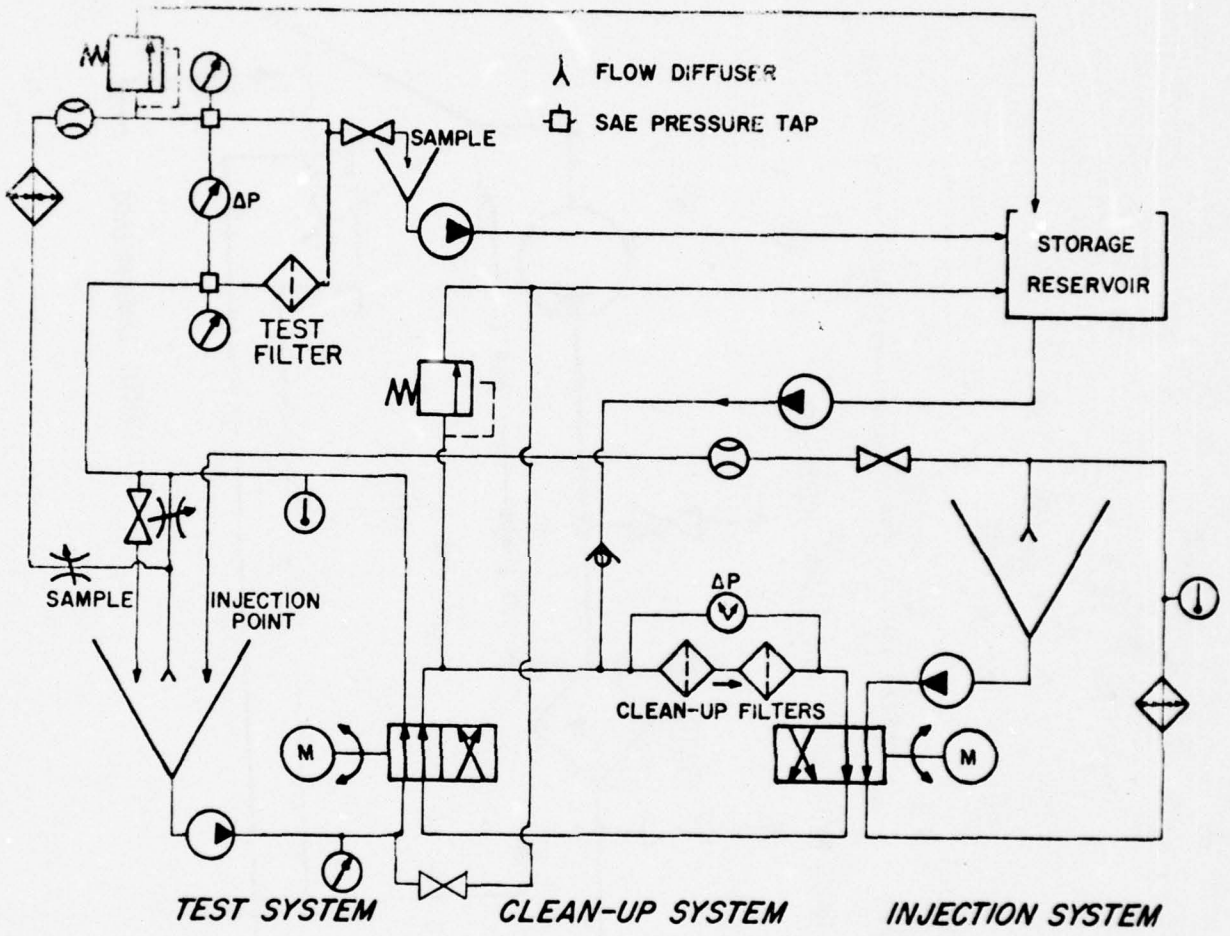


Figure A-2. Detailed Multi-Pass Test Circuit Schematic

function of the clean-up system was to clean the system fluid to the required contamination level prior to conducting a test. The clean-up filters utilized had an average filtration ratio of 10 micrometres in excess of 500 which corresponds to a cumulative efficiency of 99.8%. This high filtration ratio resulted in shorter clean-up times between tests.

A transparent plastic enclosure was constructed to completely house the multi-pass test facility as well as the electrostatic measurement apparatus. This enclosure allowed the environmental conditions surrounding the test facility to be accurately controlled. Relative humidity was controlled by an automatic humidifier and de-humidifier and temperature was controlled by an air conditioner and heater as required. Relative humidity measurements were made with a Beckman "Humi-Chek" solid state relative humidity indicator as well as with a sling psychrometer. The variation between the two measurement devices was generally less than 10%; however, the calibrated Beckman instrument readings were reported in the data.

APPENDIX B

MULTI-PASS HYDRAULIC FILTER TEST PROCEUDRE

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The standard multi-pass test method [27] utilizes the test facility described in Appendix A for evaluating the filtration performance characteristics of a hydraulic filter. The name "multi-pass" was derived from the most characteristic feature of the test — the recirculation of the contaminant which permeates through the filter. In addition to the "multi-passed" contaminant, a fresh amount of contaminant is continually introduced into the reservoir from the injection system. The contaminant injection is continued until the filter pressure differential has increased above a predetermined level called the terminal pressure drop.

The multi-pass test was developed in order to simulate an actual operating environment as closely as possible while still maintaining repeatability and reproducibility characteristics. AC Fine Test dust (commercially available from the AC Spark Plug Division of General Motors Corporation) is utilized as the test contaminant. This test dust was chosen to simulate actual contaminant. This test dust was chosen to simulate actual contaminants found in operating systems and because of the consistency of the dust from batch to batch.

The contaminant injection is made in a slurry form at a flow rate generally in the range of 0.25 - 0.50 ml/min. The injection rate expressed in milli-grams/minute is determined such that the theoretical contamination level upstream of the filter is 10 mg/l. The 10 mg/l level is an arbitrary number selected to improve test

repeatability. The injection rate can be calculated as follows:

Injection Rate (mg/min) = (10 mg/l) x (Filter flow rate, l/min)

The volume of fluid (MIL-H-5606 hydraulic fluid) in the test system is set equal to one-fourth the filter flow rate (per minute) value. Thus, a 40 l/min filter would be tested with a total circulating system volume of 10 litres. A continuous sample is removed from the system at a point downstream of the test filter. This downstream sample flow rate is set equal to the injection flow rate, thus the volume of fluid in the test circuit is maintained constant. Numerous tests as well as mathematical analyses have shown that the contaminant removed by the downstream sample is insignificant for all practical test conditions and does not alter the test results.

During a multi-pass test, fluid samples are extracted from a turbulent region both upstream and downstream of the test filter. The sample times are determined by the pressure loading characteristics of the filter such that they occur at designated predetermined pressure drop values. This helps to insure test repeatability. During the research effort reported in this dissertation additional samples were also occasionally taken at predetermined intervals in order to allow a more complete description of the filter performance throughout the life of the filter.

The fluid samples from a multi-pass test are analyzed for particle size distribution with an automatic particle counter calibrated with AC Fine Test Dust per [32]. The particle counter utilized during this research investigation was a HIAC model PC-320 with twelve particle size ranges. All particle counts were determined on a cumulative basis at 2, 3, 5, 7, 10, 15, 20, 25, 30, 35, 40, and 50 micrometres.

The primary figure of merit for a hydraulic filter is the indicator of its ability to capture and retain particles of contaminant. In a standard multi-pass test, this parameter is the filtration ratio defined as the number of particles greater than a given particle size in the influent fluid divided by the number of particles greater than the same size in the effluent fluid. A filtration or "beta" ratio of unity signifies no particle separation while a high beta ratio means that the filter is removing a significant amount of contaminant. A beta value of two indicates a 50% cumulative efficiency as the downstream contamination level would be one-half of the upstream value.

In addition to the filtration ratio, another important figure of merit resulting from the multi-pass test is the ACFTD capacity. This is the apparent contaminant capacity of the filter when exposed to AC Fine Test Dust on a multi-pass basis. The ACFTD capacity is calculated by multiplying the injection rate (mg/min) times the test termination time (minutes) and is usually

expressed in grams. The ACFTD capacity is referred to as an apparent capacity because it represents the amount of contaminant added to the filter system before plugging and not the amount retained by the filter. An extremely poor filter will generally exhibit a high ACFTD capacity because it is removing very little contaminant. Thus, the ACFTD capacity cannot alone be a useful merit parameter but should be reported with the filtration ratios.

Some modifications to the standard multi-pass filter test method were made during this investigation as reported previously. However, the basic concepts and standardized test conditions were followed. The test results obtained as a result of this program should thus have a greater applicability to existing data from other laboratories and can be directly compared to other multi-pass test results.

APPENDIX C

ELECTRICAL MEASUREMENT TECHNIQUES

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APPENDIX C

ELECTRICAL MEASUREMENT TECHNIQUES

During the course of this investigation, it was desired to conduct measurements of several electrically associated parameters - fluid conductivity, dilectric constant and electrostatic charge density. Whenever possible, standard techniques were utilized; however, most measurements required modifications for the particular application. The following is a detailed summary of the measurement apparatus and techniques utilized in the study.

FLUID ELECTRICAL CONDUCTIVITY

The electrical conductivity of a fluid is a measure of the discharge time of a capacity which uses the fluid as a dielectric. The units of conductivity are $\frac{1}{\Omega m}$ or mho/m. Because of the magnitude of the conductivity for many hydrocarbons, it is sometimes expressed in units of picomho/m. The D. C. method was utilized in this investigation for measurement of conductivity. The procedure followed was generally in accordance with ASTM D 1169-74 [33] which is actually a method for determining the specific resistance or resistivity (inverse of conductivity) of electrical insulating liquids.

The conductivity cell utilized was a standard commercially available device manufactured by Balsbaugh Laboratories, Model

No. LRC-1. The cell has a nominal air capacitance of 50 pf, a 1.27 mm electrode spacing, and requires approximately 25 ml of fluid for measurements. The actual capacitance of the cell was measured with a Tektronix type 130 L-C meter to be 55.5 pf.

The test procedure for measuring the conductivity of the test fluids was the following:

1) Clean the test cell with petroleum ether than allow to dry.
2) Check the resistivity of the cell with air as the dielectric to insure adequate cleaning and drying.
3) Add 25 ml of liquid to be tested.
4) Apply a constant DC voltage equal to 100 V across the cell electrodes.
5) Measure and record the current after two minutes have elapsed.
6) Calculate the resistivity by dividing the measured amperage into 100 V and multiplying by the cell constant.
7) Calculate the conductivity by taking the inverse of the resistivity.

The cell constant expressed in metres was calculated in accordance with ASTM 1169 by multiplying the cell capacitance with air as the dielectric times 0.113. Thus the cell constant for the model LRC-1 cell utilized was 0.113 x 55.5 = 6.27m. The voltage and current measurement were performed with a Keithley model 610C solid state electrometer. The electrometer has an input resistance

of greater than 10^{14} ohms and an offset current of less than 5×10^{-15} amperes. A large number of tests were conducted on fluid samples with various conductivities in order to refine the test method for obtaining highest repeatability.

FLUID DIELECTRIC CONSTANT

The dielectric constant of the MIL-H-5606 hydraulic fluid was measured with the Balsbaugh reference cell utilized for the conductivity measurements. The ASTM D 924 procedure [34] was generally follwed then making the dielectric measurements. The dielectric constant of a liquid is determined by dividing the capacitance of the test cell with the liquid as the dielectric by the capacitance with air as the dielectric. The capacitances were measured with the Tektronix L-C meter. The dielectric constant for MIL-H-5606 hydraulic fluid was measured to be 1.79 and did not vary significantly under the presence of anti-static additives.

ELECTROSTATIC CHARGE DENSITY

A most important measurement taken during this study was the electrostatic charge density in the liquids. The ASTM D 2679 procedure [35] for measuring electrostatic charge was generally followed during the testing. A Faraday cage was constructed as illustrated in Fig. C-1 with two concentric stainless steel containers. The volume of the inner container utilized was 425 ml.

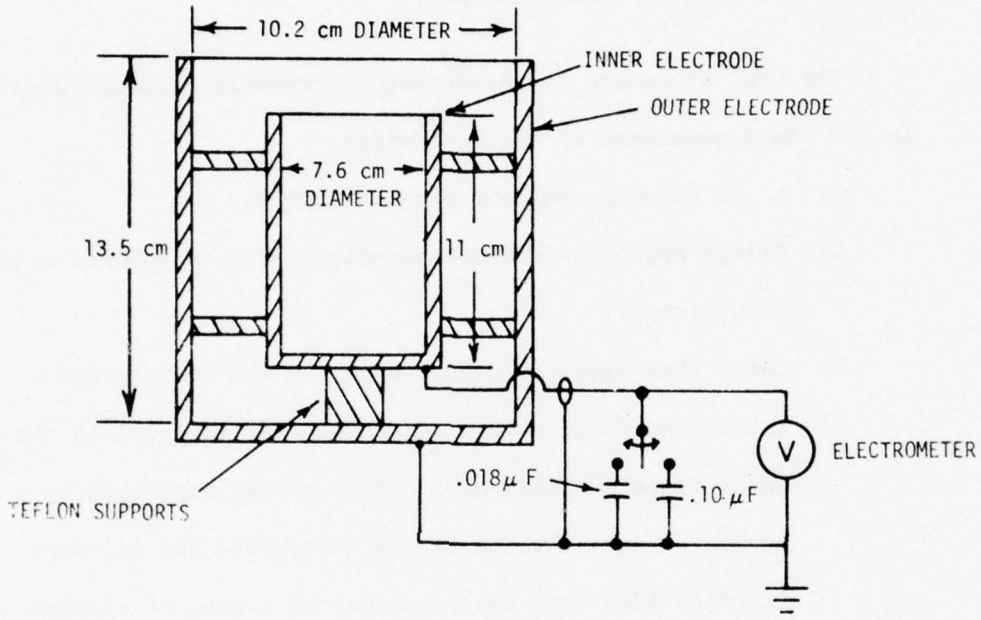


Figure C-1. Faraday Cage for Electrostatic Charge Measurements

The measured capacitance of the Faraday cage was 205 pf with the line and electrometer adding an additional 120 pf. For high charge measurements, two additional shunt capacitors were utilized with values of 0.018 and 0.10 pf.

The test procedure for measuring electrostatic charge density in the fluid consisted of the following:

1) Clean Faraday cage and dry thoroughly.
2) Select appropriate shunt capacitor is required and connect into circuit.
3) Zero, then open the electrometer in the voltage mode.
4) Insert the fluid sample to the designated level in the inner container (425 ml). (This was accomplished by opening a sample valve on the test stand and allowing fluid to flow into the container at a rate of approximately 5 l/min.)
5) Measure and record the voltage on the electrometer as soon as the sample is inside the container.
6) Calculate the electrostatic charge density by the following relationship:

$$\text{Charge Density } \frac{C}{m^3} = \frac{\text{(total capacitance, farads)} \quad \text{(voltage, volts)}}{\text{Fluid Volume } (m^3)}$$

When the relative humidity of the surrounding atmosphere was less than approximately 50%, the voltage reading was extremely stable for a period of several seconds and was independent of the fluid sample flow rate. However, with high relative humidities in the

order of 80%, the charge decay was much more rapid and accurate readings become more difficult to measure. To maintain maximum repeatability, the voltage reading was recorded as quickly as possible after the sample flow was stopped. A high sample flow rate (approximately 5 litres (minute) was also utilized to minimize charge relaxation during the sample period.

APPENDIX D

SUMMARY OF REPRESENTATIVE TEST DATA

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SUMMARY OF FILTRATION RATIOS

(PARTICLE SIZES INDICATED IN MICROMETERS)

FILTER ID.	DOWNSTREAM CHARGE (.001C/M**3)	>2	>3	>5	>7	>10	>15	>20	>25	>30	>35	>40	>50
A 1	1.87E-04	1.21	1.40	1.75	2.68	5.22	18.07	40.99	49.34	46.97	38.88	41.73	34.37
A 2	6.58E-04	1.15	1.27	1.51	2.23	4.28	16.59	48.96	79.54	81.83	73.26	50.84	36.54
A 3	8.72E-04	1.24	1.40	1.75	2.62	5.17	22.27	59.89	78.79	64.87	76.20	67.13	70.87
A 4	5.74E-03	1.09	1.17	1.36	1.86	3.24	11.16	21.36	26.03	20.18	15.86	15.83	14.26
A 5	2.55E+00	1.17	1.31	1.64	2.50	4.86	13.05	17.61	17.14	17.63	15.62	12.76	16.21
A 6	4.38E+00	1.39	1.64	2.08	3.19	6.37	23.46	48.02	47.13	43.65	40.90	37.73	36.75
A 7	1.17E+01	1.85	1.96	2.43	3.42	5.41	8.47	10.10	13.62	14.34	16.06	12.62	14.85
A 8	1.09E+01	2.67	3.38	4.63	7.62	13.22	19.39	20.15	24.11	42.72	51.76	58.57	50.75
B 1	1.45E-05	1.09	1.15	1.31	1.74	3.01	9.12	15.40	15.87	15.25	11.83	8.34	10.42
B 2	1.76E-04	1.19	1.31	1.58	2.30	4.39	14.31	24.58	28.21	27.19	25.01	23.34	20.71
B 3	2.14E-04	1.21	1.34	1.59	2.28	4.25	15.85	35.34	45.72	43.81	34.07	30.87	28.75
B 4	2.45E-03	1.08	1.15	1.31	1.77	3.19	8.43	12.83	14.39	13.84	12.56	12.48	16.01
B 5	1.87E+00	1.19	1.32	1.60	2.38	4.84	19.02	46.82	51.65	47.52	46.62	44.92	30.17
B 6	3.31E+00	1.21	1.37	1.72	2.72	5.98	31.03	99.83	117.49	143.95	104.93	89.34	40.30
B 7	7.05E+00	1.30	1.52	1.91	3.57	7.20	26.95	37.94	40.73	55.64	60.87	59.89	71.38
C 1	5.74E-04	1.04	1.05	1.06	1.09	1.15	1.44	2.22	3.93	8.58	18.81	37.91	*****
C 2	5.96E-03	1.06	1.07	1.07	1.10	1.15	1.44	2.53	4.67	11.00	29.31	27.09	33.23
C 3	1.92E+01	1.67	1.80	1.94	2.21	2.69	4.16	5.92	7.56	10.38	10.31	10.44	8.22
D 1	2.83E-04	1.19	1.43	2.15	4.34	8.36	13.33	14.28	14.72	14.78	13.87	11.99	12.61
D 2	4.21E-04	1.29	1.67	2.62	4.94	7.48	8.33	8.52	7.87	7.09	6.63	5.37	5.64
D 3	6.66E-03	1.29	1.63	2.56	5.79	16.33	40.13	45.68	48.15	43.51	44.48	42.04	32.94
D 4	7.65E-03	1.66	2.54	4.27	8.40	12.50	13.98	14.71	16.10	17.26	14.97	14.55	12.11
D 5	1.19E-02	1.33	1.73	2.98	7.52	11.56	13.67	15.06	12.70	10.37	11.34	11.14	9.76
D 6	2.93E+00	1.93	3.14	6.39	15.26	26.61	33.99	37.15	37.68	33.11	33.46	35.50	27.87
D 7	1.52E+01	3.36	5.55	9.82	18.20	27.24	29.94	32.51	32.17	33.04	32.20	33.83	29.37
E 1	1.21E-04	1.01	1.01	1.02	1.02	1.06	1.23	1.88	3.45	5.74	10.59	9.62	7.19
E 2	1.45E-04	1.03	1.03	1.04	1.07	1.13	1.36	1.96	3.46	5.67	10.43	12.96	18.17
E 3	1.11E-03	1.03	1.05	1.06	1.08	1.14	1.36	1.78	2.83	4.16	6.99	7.56	5.93
E 4	1.26E-03	1.04	1.05	1.06	1.09	1.14	1.36	1.84	2.60	3.93	5.06	5.64	5.59
E 5	7.04E-03	1.04	1.06	1.08	1.10	1.17	1.46	2.27	2.56	3.67	4.11	3.94	3.18
E 6	3.35E-01	1.06	1.07	1.10	1.14	1.23	1.59	2.68	5.04	11.84	23.20	57.36	115.46
E 7	2.92E+00	1.09	1.11	1.14	1.19	1.28	1.68	2.72	5.41	10.82	21.67	28.34	51.04
F 1	9.18E-05	2.63	5.03	11.42	23.38	27.80	29.20	52.27	68.46	88.97	93.28	112.75	186.93
F 2	7.76E-04	2.09	3.72	8.53	25.28	63.67	106.16	119.12	142.09	137.33	162.96	114.52	123.04
F 3	1.09E-03	3.08	6.16	14.37	34.61	57.68	76.23	90.80	127.16	139.83	166.60	198.79	337.08
F 4	1.15E-03	2.10	4.03	9.88	23.14	34.56	40.74	49.62	73.89	84.80	107.70	102.82	152.40
F 5	1.19E-03	2.87	5.91	13.70	45.19	147.07	348.61	379.58	394.65	536.87	461.32	395.67	329.29
F 6	3.82E-01	3.97	10.31	32.26	95.45	179.54	183.37	238.03	307.82	368.50	369.59	425.08	******
F 7	2.67E+00	4.22	9.48	21.81	48.64	75.15	91.12	128.46	154.47	143.91	107.41	103.52	169.19

FLOW RATE FOR A1 - A7 WAS 114 L/MIN.
FLOW RATE FOR OTHERS WAS 76 L/MIN.
TEST FLUID: ISO - H - 5606, 38C

SUMMARY OF AVERAGE RETENTION RATIOS

FILTER ID.	DOWNSTREAM CHARGE (.001C/M**3)	(PARTICLE SIZE INTERVALS INDICATED IN MICROMETERS) 2-3	3-5	5-7	7-10	10-15	15-20	20-25	25-30	30-35	35-40	40-50	>50
A 1	1.87E-04	1.06	1.20	1.44	2.04	3.98	12.27	34.33	57.71	68.25	32.34	52.55	34.37
A 2	6.58E-04	1.06	1.14	1.30	1.78	3.32	10.96	33.33	76.68	96.37	238.29	78.66	36.54
A 3	8.72E-04	1.09	1.21	1.45	2.00	3.89	14.39	47.36	121.29	53.23	101.26	67.12	70.87
A 4	5.74E-03	1.03	1.08	1.21	1.54	2.60	8.21	18.20	39.91	30.86	14.95	16.26	14.26
A 5	2.55E+00	1.04	1.15	1.38	1.94	3.84	10.82	18.03	14.61	22.04	26.51	10.46	16.21
A 6	4.38E+00	1.18	1.36	1.67	2.37	4.77	16.30	48.73	53.84	49.36	49.56	36.99	36.75
A 7	1.17E+01	1.72	1.67	2.03	2.74	4.71	7.27	7.95	12.84	11.49	28.38	10.46	14.85
A 8	1.09E+01	2.02	2.49	3.26	5.46	10.78	18.36	17.32	15.47	47.64	43.83	64.85	50.75
B 1	1.45E-05	1.04	1.08	1.18	1.46	2.44	6.94	14.96	17.04	27.23	33.57	7.03	10.42
B 2	1.76E-04	1.06	1.15	1.34	1.81	3.41	10.66	21.37	29.72	31.30	29.52	26.16	20.71
B 3	2.14E-04	1.08	1.18	1.36	1.82	3.29	10.86	28.31	47.00	60.65	42.56	36.59	28.75
B 4	2.45E-03	1.03	1.07	1.18	1.47	2.61	6.53	11.36	15.18	16.37	12.50	10.56	16.01
B 5	1.87E+00	1.07	1.17	1.36	1.85	3.69	12.52	42.18	56.34	51.48	51.36	80.43	30.17
B 6	3.31E+00	1.06	1.16	1.42	2.04	4.38	19.32	85.82	90.23	299.48	123.51	449.98	40.30
B 7	7.05E+00	1.09	1.27	1.47	2.71	5.45	21.58	35.21	29.40	48.97	61.65	53.14	71.38
C 1	5.74E-04	1.03	1.04	1.05	1.06	1.11	1.27	1.82	2.27	5.09	9.77	18.41	*****
C 2	5.96E-03	1.05	1.06	1.06	1.08	1.10	1.23	1.93	2.90	5.85	33.85	22.31	33.23
C 3	1.92E+01	1.50	1.64	1.75	1.96	2.34	3.52	5.01	5.69	10.40	11.02	12.32	8.22
D 1	2.83E-04	1.06	1.21	1.67	3.18	7.01	12.54	13.62	14.55	15.62	19.82	11.41	12.61
D 2	4.21E-04	1.08	1.34	1.99	3.86	7.06	8.16	9.43	9.21	7.73	10.16	5.14	5.64
D 3	6.66E-03	1.04	1.25	1.83	3.86	12.50	36.43	43.36	56.10	42.85	48.26	55.78	32.94
D 4	7.65E-03	1.20	1.83	2.99	6.44	11.79	13.38	13.31	14.70	22.33	15.47	17.35	12.11
D 5	1.19E-02	1.07	1.29	2.13	5.82	10.66	11.85	17.51	15.00	5.81	11.72	12.63	9.76
D 6	2.93E+00	1.36	2.06	4.14	10.88	23.71	31.64	36.69	48.33	31.32	30.39	45.10	27.89
D 7	1.52E+01	2.13	3.52	6.63	13.81	25.93	27.99	32.88	30.84	34.63	29.39	36.62	29.37
E 1	1.21E-04	1.00	1.01	1.01	1.01	1.03	1.11	1.50	2.52	3.69	11.55	9.47	7.19
E 2	1.45E-04	1.01	1.02	1.03	1.04	1.10	1.23	1.55	2.37	3.31	6.64	9.59	18.17
E 3	1.11E-03	1.00	1.04	1.04	1.06	1.11	1.25	1.45	2.16	2.60	6.01	9.55	5.93
E 4	1.26E-03	1.02	1.04	1.04	1.06	1.10	1.25	1.59	2.00	3.11	4.59	5.63	5.59
E 5	7.04E-03	1.03	1.04	1.06	1.08	1.12	1.29	2.09	1.89	2.73	3.81	5.38	3.18
E 6	3.35E-01	1.03	1.05	1.07	1.10	1.17	1.36	2.04	3.10	6.57	12.90	41.80	115.46
E 7	2.92E+00	1.04	1.09	1.11	1.15	1.22	1.46	2.01	3.45	6.44	16.06	20.90	51.04
F 1	9.18E-05	1.54	2.92	7.42	20.27	27.10	21.51	42.08	53.08	81.12	75.84	83.06	186.93
F 2	7.76E-04	1.28	2.21	5.13	16.14	52.28	97.25	101.34	148.65	113.40	566.86	108.42	123.04
F 3	1.09E-03	1.73	3.53	8.85	24.87	50.60	66.72	68.07	108.96	110.72	128.57	141.27	337.08
F 4	1.15E-03	1.27	2.35	6.20	17.57	31.92	35.34	36.13	62.31	62.29	119.76	82.26	152.40
F 5	1.19E-03	1.45	3.48	7.90	27.06	110.21	324.62	364.14	274.13	742.78	403.44	1028.46	329.29
F 6	3.82E-01	1.96	5.19	18.53	65.45	177.66	154.11	191.48	249.11	373.23	300.31	225.77	******
F 7	1.67E+00	2.25	5.25	13.61	36.38	68.73	73.21	109.18	174.61	306.60	115.55	73.59	149.19

FLOW RATE FOR A1 - A7 WAS 114 L/MIN.
FLOW RATE FOR OTHERS WAS 76 L/MIN.
TEST FLUID: MIL - H - 5606, 38C

SUMMARY OF FILTRATION RATIOS

FILTER ID.	RELATIVE HUM.(%)	(PARTICLE SIZES INDICATED IN MICROMETERS) >2	>3	>5	>7	>10	>15	>20	>25	>30	>35	>40	>50
AA 1	32	1.28	1.44	1.81	2.57	4.72	11.73	17.87	19.17	22.05	19.30	22.86	20.58
AA 2	32	1.23	1.36	1.63	2.15	3.50	7.11	8.85	9.49	10.30	10.76	8.81	9.10
AA 3	42	1.23	1.39	1.74	2.38	4.00	8.37	10.09	10.80	10.91	11.44	13.96	25.53
AA 4	42	1.22	1.37	1.70	2.30	3.89	8.15	10.30	10.88	10.54	11.13	10.99	8.40
AA 5	58	1.24	1.40	1.72	2.32	3.99	8.22	10.46	12.73	11.87	11.27	10.21	8.69
AA 6	58	1.19	1.36	1.74	2.39	4.28	8.67	9.85	12.26	10.11	11.00	9.11	6.95
AA 7	70	1.16	1.28	1.55	2.28	4.36	19.71	106.18	271.81	215.78	260.53	245.05	266.58
AA 8	70	1.18	1.31	1.60	2.38	4.60	19.51	65.94	104.32	132.83	117.22	161.20	********

END OF DATA
E

SECTION VIII

HYDRAULIC FILTER QUALITY CONTROL

PROJECT PERSONNEL

Dr. E. C. Fitch, Program Director

Dr. L. E. Bensch, Program Manager

Mr. E. J. Eskilson, Project Engineer

Mr. L. C. Moore, Particle Metrologist

PREFACE

This section presents a detailed account of the project activities in the area of filter element quality control. Results of two industrial questionnaires are presented as well as two industrial advisory meetings. Finally, an accelerated multi-pass filter performance test is presented for evaluation of filter element quality control. Results of a limited testing program are also included.

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CHAPTER I

INTRODUCTION

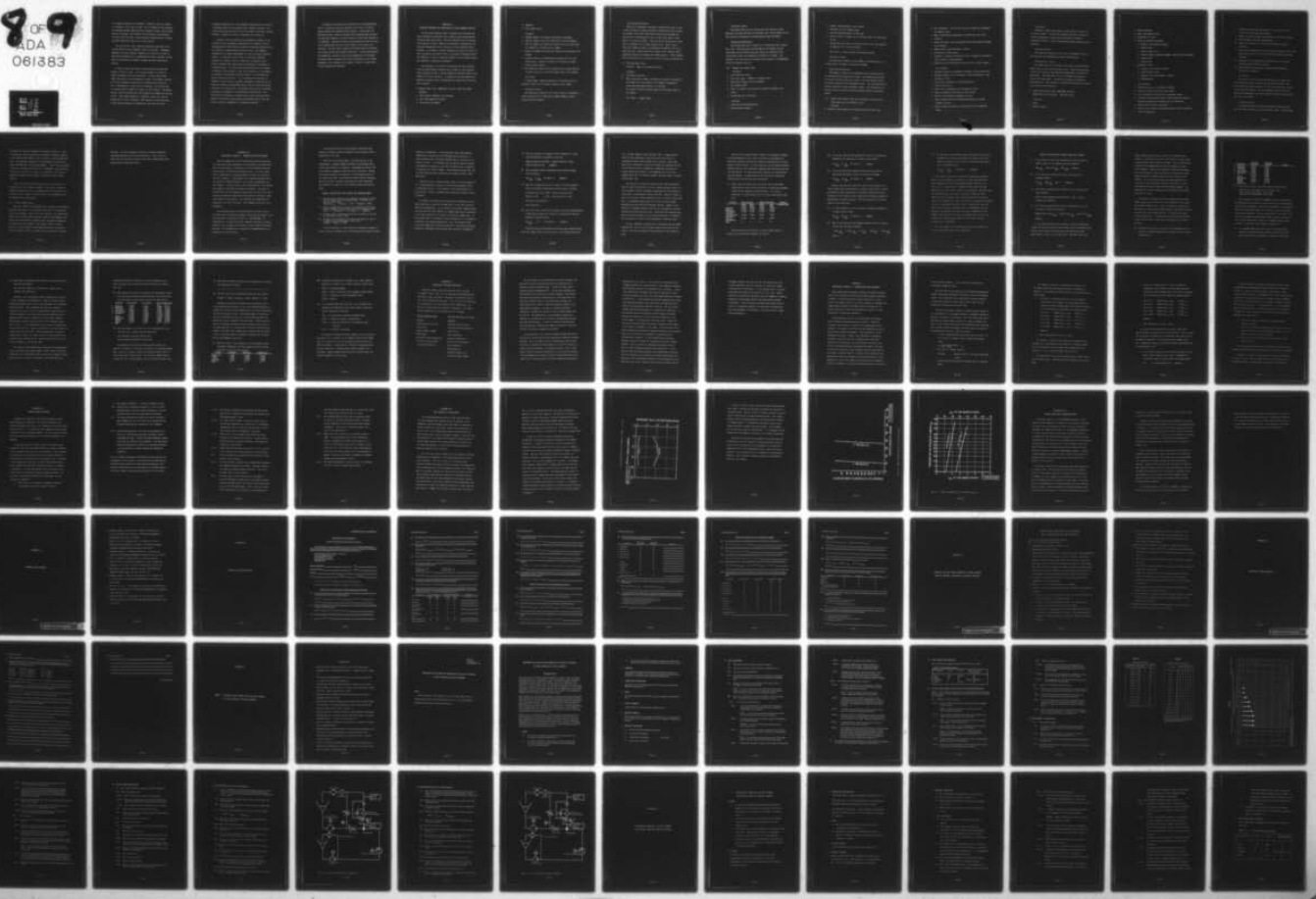
Hydraulic systems are under constant attack by contaminant throughout the life of the system. It has been demonstrated by the FPRC and other groups that contaminant is unavoidably present, to varying extents, even before the startup of a system. When the unit is put into service in a contaminated environment, ingression through rod seals, rotary seals and breathers may become significant—the rate at which contaminant is ingressed often increasing as the system is used. These types of ingression are currently being studied and, as a result, ingression rates are being decreased, but ingression of contaminant has not been eliminated.

Contaminant is, by definition, harmful and has been shown to affect all hydraulic components to varying degrees. Pumps are generally considered one of the most contaminant sensitive portions of the typical hydraulic system. As a pump wears, it requires more power to produce the same output, wasting energy that must be dissipated as heat, sometimes overtaxing the available cooling capacity. Hydraulic cylinders are also subject to significant damage by contaminant, resulting in various leakages. Leakage within the hydraulic cylinder allows "creeping" under load and contributes to system inefficiencies in the same way a worn pump does. External hydraulic cylinder leakage, with subsequent fluid losses, is often accompanied

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by increased ingression of contaminant. Hydraulic valves are subject to contaminant in at least two ways. While contaminated fluid degrades the internal sealing characteristics of hydraulic valves, they are also subject to silting. This buildup of fine particles, precipitated by the fluid within the valve, causes otherwise workable valves to require excessive operational forces or to sieze entirely.

The rate at which these components degrade has been shown to be closely related to the contaminant level of the system. Contaminant degradation of other hydraulic components, at this point in time, is generally overshadowed by the components listed above, although, in some cases, other portions of hydraulic systems have been significantly affected.

In order to limit wear rates of various hydraulic system components, a filter is used. It is the sole purpose of a filter to remove contaminants from the system fluid. An ideal filter would remove 100 percent of the contaminant to which it was exposed, and have an infinite contaminant capacity with no pressure drop across the unit. Unfortunately, no such filter is available. The filters that are available can be judged with respect to their ideality. Efficiency, capacity, and the pressure drop can be measured and compared for various filters. These three parameters, in addition to unit cost, must be weighed when a filter is required. The efficiency of the filter must be high enough to maintain a contamination level that will yield

acceptable component life. The contaminant capacity must be such that a reasonable amount of time between filter changes is allowed. The pressure drop characteristics of the filter must be such that the power required to operate the filter is compatible with the system.

Currently there are industrial and government standards for the evaluation of the performance characteristics of hydraulic filters. Existing military specifications are based on Qualified Products Lists. The presence of a manufacturer's product on such a list suggests confidence in the product performance, but does not assure consistent quality. At this point in time, there is no military standard, acceptable to industry, that will insure that a hydraulic filter element purchased after qualification will perform in conformance to the specification. The current military buying practice with regard to hydraulic filter elements is such that the majority of the elements so purchased are considered "military specials" by the manufacturer and are often priced as such. It is the purpose of this project to formulate a quality control procedure, acceptable to manufacturers and usable by military and nonmilitary buyers, that will allow the quality of filters to be monitored on a statistical basis and allow reasonable confidence of conformance to the appropriate specification. This should allow the military to procure quality filters from any vendor capable of producing a worthy product and eliminate much of the need for the filters so purchased to be considered "specials."

The goals of the project were fulfilled in the following manner. First, a literature survey was conducted in an effort to discover present methods of determining filter quality. A survey was then prepared and sent to many of the major filter manufacturers and users. The excellent response to the survey allowed a first draft proposal to be formulated. Industrial advisory group meetings were held with both filter manufacturers and users and the goals of the project were presented. The first draft proposal was presented and the discussion that ensued yielded some valuable information. A second questionnaire was formulated, covering critical topics suggested at the meetings. Testing was done to explore some of the distributions of the various parameters to be included in the standard. Finally, a second draft of the procedure was formulated.

CHAPTER II

EXISTING METHODS FOR MONITORING FILTER ELEMENT QUALITY

The Fluid Power Research Center, Oklahoma State University has tested thousands of hydraulic filter elements using various methods during the past twenty years. This past experience with the testing and evaluation of filter elements was considered a necessary but not sufficient background for the creation of a filter quality control standard. In an effort to further determine the "state of the art" in filter evaluation and quality control, an extensive literature survey was conducted. A list of references related to filter quality control and evaluation is presented in Appendix A.

Past experience at the FPRC in conjunction with the literature survey and questionnaire indicated that a number of tests were generally accepted and used to various degrees by different groups in the industry for filter evaluation. A synopsis of the more popular test methods is presented below to familiarize the reader with the various tests available.

I. Bubble Point Test (ANSI/B93.22-1972, ISO 2942-1974)

Equipment:

1. Test reservoir (similar to an aquarium).
2. Test fluid (generally alcohol).
3. Low pressure air supply.

4. Manometer.
5. Test element holder.

Procedure:

1. Immerse element to be tested, saturating it thoroughly.
2. Seal the element in the test holder while both are immersed.
3. With the element just below the surface of the test fluid apply air pressure to the inside of the element.
4. Rotate the element just below the surface while watching for air bubbles.
5. Continue rotation at increasing air pressure until the first bubble occurs. Usually this bubble is followed by a steady stream of small bubbles.
6. The pressure at which the first bubble occurs, usually measured in inches of water, is called the bubble point of the element.

Data Format:

A single number, sometimes presented in conjunction with the location at which the first bubble occurred on the element.

Information Gained:

The fabrication integrity of the filter element is determined. A poorly sealed end cap or side seam or element damage is easily detected with this method.

Advantages/Disadvantages:

While this is generally considered a nondestructive test in that it does not harm the element (provided a suitable test fluid is used), the presence of the test fluid on and within the test specimen has been noted as a disadvantage and has, in some cases, made it uneconomical to retain the element. This test has not been shown to relate well to efficiency for a fibrous pleated filter. While a poorly fabricated element may have a low separation efficiency, this is not always the case. Bubble point has no affect on dirt capacity and no significant relation to pressure drop characteristics. Filter elements that are enclosed, such as spin-ons, are very difficult to bubble test.

II. Boiling Point Test

Equipment: Same as for bubble point test.

Procedure:

1-3. From bubble point test.

4. While rotating the element, increasing air pressure is applied to the element until the surface of the element is passing air across the entire media surface, or "boiling."

5. The pressure at which "boiling" began is the "boiling point" of the element.

Data Format: A single number.

Information Gained:

Some groups feel that the boiling point test assists in determination of the mean pore size for the media used. Non-uniformity of air flow could be an indication of discontinuities within the media.

Advantages/Disadvantages:

The boiling point test is easily run in conjunction with a bubble point test. This method has the advantage that the higher air flow associated with the boiling point test allows the condition of the entire media surface to be inspected, provided it is visible. The presence of the test fluid on the filter after testing is a disadvantage, as with the bubble point test.

III. Tracer Gas (Sock) Test

Equipment:

1. Low pressure air supply.
2. Tracer gas supply. (Ammonia is commonly used.)
3. Ultraviolet lighting for test specimen.
4. Test element holder.
5. "Sock," an elastic cloth sleeve to surround the element to be tested.
6. Fluorescing dye to treat sock.

Procedure:

1. Treat sock with fluorescing dye.
2. Place sock over element.

3. Install covered element in test fixture.

4. Establish air flow through element.

5. Introduce measured amount of tracer gas.

6. Monitor surface of element with ultraviolet light for fluorescing of dye.

7. Fluorescing of the dye in a particular place on the sock indicates an imperfection in the filter media.

Data Format: Usually pass/fail evaluations are made.

Information Gained:

The fabrication integrity of the element is determined and, to a certain extent, the media is inspected for damage.

Advantages/Disadvantages:

The element is ready for use immediately after evaluation and is not saturated with any special test fluid. The fluorescing dyes used in the past have been difficult to apply to the sock uniformly and in the proper concentration for dependable results. The trace gas must be precisely metered to obtain repeatable results. The fluorescence is a transient feature that must be observed during the short fluorescing period.

IV. Single Pass Filter Test (The following is typical of the many single pass test methods in use.)

Equipment:

1. Reservoirs large enough to hold sufficient fluid for the test.

2. Test contaminant. (AC Test Dust as well as spherical contaminants are commonly used.)
3. Means of maintaining contaminant in suspension in test fluid reservoir.
4. Steady flow pump to move contaminated fluid from reservoir through test specimens.
5. Sample points up and downstream of filter.
6. Differential pressure gauge.
7. Equipment to evaluate contamination level of samples by gravimetric and/or particle counting methods.
8. Sample containers sufficiently clean so as not to affect samples.

Test Procedure:

1. Determine quantity of contaminant necessary to fully load filter.
2. Determine quantity of oil necessary to carry contaminant at some gravimetric level.
3. Mix oil and contaminant in upstream reservoir.
4. Install test filter.
5. Start flow of contaminated oil through test filter.
6. Sample as necessary, according to specification.
7. Maintain flow uniformly throughout the test.
8. When the specified filter differential pressure is reached, terminate the test.
9. Analyze samples according to specified method for contaminant levels.

Data Format:

Generally a number indicating the single pass dirt capacity of the filter element. If particulate analysis was done, the particle counts and gravimetric levels would be available to indicate separation efficiency. Time vs differential pressure data may also be available.

Information Gained:

The single pass capacity of the filter is determined as well as the efficiency and differential pressure characteristics.

Advantages/Disadvantages

The single pass test yields data on the three vital filter parameters, capacity, efficiency and pressure drop. There are many versions of this test, each with advantages and disadvantages in any specific application. The single pass test is not designed to simulate an actual fluid power system. Correlation of single pass capacities, and efficiencies, those observed in the multi-pass situation of a "real life" system is often difficult.

V. Multi-Pass Filter Test (ANSI/B93.31-1973, NFPA Std T3.10.8.8-1973, ISO DIS 4572)

Equipment:

1. Timer
2. Particle counter

3. ACFTD contaminant

4. Super clean sample bottles

5. Mil-H-5606 test fluid

6. Filter test system containing:

   a. Reservoir per reference standard

   b. Hydraulic pump

   c. System cleanup filter

   d. Pressure gauges, temperature gauges, controller and flow meter

   e. Pressure taps

   f. Sample points

7. Contaminant system

   a. Reservoir per reference standard

   b. System cleanup filter

   c. Hydraulic pump (centrifugal or similar)

   d. Sampling means

Test Procedure:

1. Determine quantity of contaminant necessary.

2. Determine required injection system volume.

3. Mix contaminant with clean oil in contaminant system.

4. Determine housing pressure drop by installing empty housing in system and operating at specified flow and temperature.

5. Clean up fluid in filter test system.

6. Install filter element and determine pressure drop at specified flow and temperature.

7. Calculate pressure drop increases of 5, 10, 20, 40, 80, 100 percent of net pressure drop allowed.
8. Adjust test system volume to one quarter of the flow per minute.
9. Start injection and timer simultaneously.
10. Record test time for 5, 10, 20 40, 80, 100 percent pressure increases.
11. Sample up and downstream of the filter after 2 minutes (only in the ISO version) and at 10, 20, 40, 80 percent differential pressure increases.
12. Terminate test by stopping flow through the test filter when terminal pressure (100%) is reached.
13. Analyze samples using particle counting and/or gravimetric techniques, as required.

Data Format:

Raw particle counts are generally reported as the quantity of particles greater than a given size per millilitre of fluid. Particle counts are generally made at 10, 20, 30, and 40 microns, for all up and downstream samples. Also the gravimetric levels of the 80 percent upstream sample, and of the injection system, before and after the test, are reported.

Information Gained:

A standard multi-pass test yields information relating to how the filter will perform in an actual hydraulic system. The multi-pass

test reports on all the vital parameters of the filter, the efficiency and/or filtration ratio, the contaminant capacity, and the pressure loss across the filter. Information about filtration ratio and other parameters are reported at a number of points during the "life" of the element.

Advantages/Disadvantages:

A multi-pass test stand is a complex testing device, and may represent a significant capital investment. Equipment for analysis of samples taken during the test is also rather sophisticated. A multi-pass test can be applied to a wide range of flow rates, pressure drops, and contaminant capacities using the same basic test stand. The multi-passing action of the test with constant ingression was selected to simulate the "typical" hydraulic system and contaminant not removed by the filter remains in circulation. This closely resembles the operation of a filter in an actual hydraulic system. Multi-pass data have been used to optimize filter-component costs and give a true picture of what contaminant levels can be expected with a specific filter and a given contaminant ingression rate. This can be used to predict the life of the system and necessary frequency of service.

VI. Element Manufacturing Process Q. C. Checks

There are a number of checks that can be performed during the manufacturing process in order to insure that the elements produced are the proper size, have no sharp edges, are fabricated from specified

materials, are properly assembled, are properly sealed, etc. Each of these tests requires some specific procedure, generally composed by the manufacturing company's quality control or engineering department. The equipment necessary for the performance of these tests varies depending upon the measured parameter. These tests allow manufacturers to determine whether or not a subassembly is or will cause the filter to be defective, before it is completed. This allows a significant reduction in the number of defective units of final product.

These tests allow some of the properties of the filter to be evaluated before the element is complete. The buyer is concerned primarily with whether or not the completed unit is acceptable. In-process checks for quality are often useful to the manufacturer but are generally kept confidential.

VII. Filter Media Tests

There are a number of standardized tests available for evaluating paper and other types of filter media, before they are incorporated into a filter element. Some of the tests used are: air permeability, bubble point, boiling point, bead tests, capacity and plugging tests, tensile test, Mullen burst, stiffness tests, corrugation dimensions (if applicable), basis weight, resin content, volatile content, Frazier air flow, hydraulic fluid compatibility tests, and others. The entire battery of tests is seldom run on any given lot of media.

The tests to be run are generally selected to evaluate properties considered important by the filter manufacturer. Few, if any, of these tests can be directly related to the three primary filter parameters, based on media tests alone.

CHAPTER III

INDUSTRIAL SURVEY I, FORMULATION AND RESPONSE

Once the common tests for the determination and for monitoring of filter quality were assembled, it was necessary to determine which tests were acceptable to the majority of filter manufacturers/users and the extent to which they are presently used. In order to make such a determination, it became necessary to survey the opinions of the filter industry. A questionnaire was formulated (Appendix B) by the staff of the FPRC and mailed to 126 persons, representing over 100 companies. Forty-two (33%) of these questionnaires were returned; and a number of individuals responded by telephone. The staff of the FPRC felt that this impressive response was due to the fact that a military standard for monitoring filter quality could have a significant impact on the industry. The responses were summarized and analyzed by the staff of the FPRC. Brief summaries were sent to all respondents.

The questionnaire had four distinct parts, the first portion included a short introductory paragraph, discussing the purpose of the survey and OSU's association with the U.S. Army MERADCOM. The respondent was to provide his and the company's name and to indicate whether or not he wished to participate in future industrial advisory meetings. The respondent was to classify his company/division as a manufacturer or end-item user.

The second section of the Questionnaire, delineated below consisted of general questions intended for both hydraulic filter manufacturers and users.

Below are the questions asked, in the same form as on the questionnaire. Response summaries are shown in parentheses where a blank was provided for a response. Percentages are based on the total number of responses, except where noted otherwise. Because not everyone answered every question, the percentages given will not always total 100%. Some questions allowed multiple responses, such as G7 and G9, on these the percentage of those responding who approved of or made use of the various concepts/procedures is shown.

GENERAL SECTION FOR BOTH USERS AND MANUFACTURERS

G1. How many types (differenct part numbers) of hydraulic filters does you company deal with (manufacturer or receive) on a typical business day?
Minimum no.____ Average no. (2-10) Maximum no. (200+)

G2. Would the manufacture or receipt of any given type of filter element normally be considered a batch or continuous process?
Batch ¤ Continuous ¤ No opinion ¤ Comments:

G3. How many types of hydraulic filters do you produce or buy only on a speciality basis? (0-60)

G4. How many types of hydraulic filters do you produce or buy on a regular or continuous basis? (10-15 ave.)
Comments: (Range 1 to 100+)

The first question, G1, was intended to determine the number of types of filters that might need to be inspected by the quality control

method to be formulated. It was found that, while some companies manufacture or receive over 100 individual types of filters on a typical business day, the majority manufactured or received 2 to 10 types of filters. It is apparent that the quality control procedure will generally be applied to less than 10 types per day, in general, if applied to all filter elements. (The possibility exists that lube oil filters may have been included in this number by some of those questioned.)

Question G2 was to indicate whether, under the current method of Q.C., the manufacture or receipt of filters was considered a batch or continuous process. It was classed by over three-quarters as a batch process. The Q.C. procedure should therefore be keyed to batch evaluation.

The number of specialty filters produced or purchased by any one group, Question G3, was found to range from 0 to 60, with the lesser quantities predominating. It was reported that several companies dealt with over 200 filter part numbers on a regular basis while the majority of the trade involved 10 to 15 types of filters per company (G4). It would appear that if all hydraulic filter elements are to be evaluated, according to the proposed standard Q.C. test, that as many as 200 filter part numbers will require specification. Production quantities, ratings and requirements were surveyed with questions G5 through G8.

G5. What is, typically, the number of filter elements of a given type manufactured or purchased at one time?

Highest production (50,000) Average production (1500)

Lowest production (50) Comments:

G6. Do your quality control requirements vary from one element type to another?

Yes ¤ (43%) No ¤ (40%) No opinion ¤ Comments:

G7. What type of method do you use to refer to a filter element's particle separation characteristics when buying or selling?

Filtration ratio (β) ¤ (58%) Absolute rating ¤ (12%)

Nominal rating ¤ (64%) Mean pore size ¤ (15%)

Efficiency (how measured)

Other (describe)

G8. Do you feel that a valid filter quality control test method should produce results which can be directly correlated with particle separation characteristics?

Yes ¤ (89%) No ¤ No opinion ¤ Comments:

Question G5 gave an indication of the batch sizes currently being used, they ranged from as few as 50 units to over 50,000 filters per

lot. The most common lot size was about 1500. A slight majority (54%) of those responding to Question G6 felt that their Q.C. requirements varied from one element type to another. While this question was intended to contrast Q.C. methods for hydraulic filters for various uses (such as hydrostatic transmission filtration vs backhoe return line filters), it was apparent from a few of the comments given that the different element types were taken to include varying media types.

The response to Question G7 indicated that, while the majority (64%) of filter buyers/users still use a nominal rating to refer to a filter element's particle separation characteristics, the filtration ratio, Beta, is becoming more widely used (58%). Absolute ratings are used by 12 percent of those responding, while 15 percent use mean pore size. An individual group may use two or more of these ratings depending upon the persons and companies involved. Comments accompanying the responses indicated that the use of filtration ratio was increasing and was preferred by most. Discussion also indicated that there are various opinions as to the proper definition of "absolute" and the method to be used in the determination of mean pore size.

Of those responding to Question G8, a significant majority (89%) indicated that they felt that a filter quality control test should relate to the particle separation characteristics of the element involved.

Question G9 is presented below, showing the percentage acceptance of each nondestructive test listed in relation to determination of fabrication integrity and particle separation characteristics for both groups of filters listed. It can be seen that the only widely accepted nondestructive test for fabrication integrity was the bubble point test with 79% approval. No other fabrication integrity test received more than 25% approval. No nondestructive test for particle separation characteristics obtained more than 25% approval.

G9. Please check the blanks below if you feel that the following nondestructive tests are valid indicators of fabrication integrity and/or particle separation characteristics. If a test is applicable to a given special type of filter, please specify that type.

Test Method	Fabrication Integrity		Separation Characteristics		Comments
	All Filter Types	Special Types	All Filter Types	Special Types	(List Special Types)
Bubble Point	☐ (79%)	☐ (33%)	☐ (24%)	☐ (15%)	
Boiling Point	☐ (12%)	☐ (3%)	☐ (3%)	☐ (6%)	
Air Permeability (ΔP vs flow)	☐ (24%)	☐ (3%)	☐ (21%)	☐ (3%)	
Liquid Permeability (ΔP vs flow)	☐ (24%)	☐ (0%)	☐ (9%)	☐ (6%)	
Tracer Gas (ammonia/sock)	☐ (12%)	☐ (3%)	☐ (3%)	☐ (0%)	
Others	☐	☐	☐	☐	

Industrial opinion with respect to quality control and dirt capacity was surveyed with Questions G10, and G11.

G10. Do you feel that the dirt capacity of a filter is an important parameter to be monitored in a quality control check?

Yes ¤ (55%) No ¤ (38%) No opinion ¤ Comments:

G11. Do you feel that minor capacity variations can be noted by monitoring the number of pleats in a pleated cartridge?

Yes ¤ (54%) No ¤ (38%) No opinion ¤ Comments:

Slightly more than half (55%) felt that the dirt capacity of a filter element should be monitored on a quality control basis. Nearly two-thirds of those responding to the question indicated that minor variations in dirt capacity could be monitored by noting the number of pleats within a pleated cartridge. The final questions of the first portion of the survey were:

G12. Is clean element ΔP (in a test liquid) an important parameter in a quality control check?

Yes ¤ (61%) No ¤ (30%) No opinion ¤ Comments:

G13. What is your opinion of the maximum allowable time for a quality control test on a filter element?

<1 min. ¤ (11%) 1-5 min. ¤ (26%) 5-10 ¤ (6%) 10-15 ¤ (4%) >15 min. ¤ (9%)

Other ¤

G14. If a procedure could be devised that would expose a filter to a contaminant for separation appraisal then remove all the contaminants, could the filter element be considered "new" and be sold or purchased as such?

Yes ¤ (23%) No ¤ (61%) No opinion ¤ Comments:

Clean element differential pressure was considered an important quality control parameter by 61 percent of those responding (G12). Most respondents felt that the maximum allowable time for a Q.C. test was 1 to 5 minutes (G13). Comments indicated that this was appropriate for their present quality control methods. The FPRC staff was considering a test that could be applied to small lots and/or high priced filter elements that would allow an actual filtration ratio to be determined yet allow the filter to be cleaned by dissolving the test contaminant and reused as "new." Slightly more than half felt that any procedure exposing a filter to contaminant, then removing such contaminant would make the filter unsaleable as "new." Comments indicated that the chance of removing all of the test contaminant was considered slim, despite the fact that a specific procedure was not mentioned.

The third segment of the Industrial Survey was intended exclusively for hydraulic filter users and buyers.

SPECIAL SECTION FOR FILTER USERS AND BUYERS

U1. Do you currently find filter manufacturers specifying quality control limits for the elements they are marketing?

Most ¤ (23%) Only a few ¤ (27%) None ¤ (35%) Comments:

U2. Do you currently specify given limits of the quality of incoming elements?

All ¤ (33%) Some ¤ (43%) None ¤ Comments:

U3. What limits of maximum percent defective do you allow for purchased elements?

Most critical elements (0.25% defective) Least critical elements (4% defective)

U4. Do you accept manufacturers' data for establishing Q.C. or do you plan to conduct your own tests?

Manufacturers' tests ¤ (47%) Conduct tests ¤ (24%) Plan to test ¤ (29%)

Comments:

It was discovered that about one-quarter of those buying filter elements are given the quality control limits used by the manufacturer, in most cases; another 27 percent found that manufacturers were specifying Q.C. limits in a few cases (U1). Currently, 33% of the

buyers surveyed specify quality limits on all incoming filter elements. An additional 43% specify such limits on at least some of the hydraulic filter elements purchased. Very few buyers/users disclosed their maximum percent defective allowable for their filter elements. Those who did respond to Question U3 indicated that they allowed 0.25 percent defective for their more critical elements and about 4 percent defective for the least critical elements. It was found that, under the present systems of Q.C., 24 percent of the buyers/users conducted their own Q.C. tests, while another 29% plan to begin testing incoming elements. The remaining 47% of those responding were satisfied with their manufacturers Q.C. tests.

The response to Question U5, below, indicates that the most common methods used or specified for the evaluation of the quality of purchased elements were, in order of preference: bubble point, multi-pass test using Beta and ACFTD capacity, differential pressure vs liquid flow, and a single pass test. This was for the most critical applications. A single pass test is generally used when a less critical filter is evaluated, in preference to any other procedure.

U5. What method do you use or specify for the evaluation of the quality of purchased elements for your most and least critical applications?

	Test Method	Most Critical Application	Least Critical Application	Comments
a.	Bubble point	☐ (53%)	☐ (7%)	
b.	Boiling point	☐ (0%)	☐ (7%)	
c.	ΔP vs air flow	☐ (0%)	☐ (0%)	
d.	ΔP vs liquid flow	☐ (40%)	☐ (0%)	
e.	Multi-pass test	☐ (47%)	☐ (0%)	
	β	☐ (47%)	☐ (0%)	
	Capacity	☐ (53%)	☐ (0%)	
f.	Single-pass test	☐ (13%)	☐ (13%)	
	Efficiency	☐ (20%)	☐ (13%)	
	Capacity	☐ (20%)	☐ (13%)	
g.	Other(specify)	☐	☐	

U6. What are your typical costs for quality control testing expressed as a percentage of total element cost?

Most critical application: 50% of cost
Least critical application: 1% of cost

Poor response to Question U6 suggested that those filling out the questionnaire were unsure of the costs involved in their present quality control program(s). From the data available, it is suspected that filter buyers spend about 1% of the cost of a lot of elements inspecting the elements. One user reported a Q.C. cost equal to the cost of the elements purchased—these filters were to be used on a very critical application. Desire for a standard quality control technique is indicated by the response to Question U7.

U7. If a standard (NFPA, SAE, ANSI, ISO, etc.) procedure were developed for measuring filter element quality control, how would your purchase and acceptance techniques be affected?

(40%) ¤ a. Would start conducting standard tests and specifying limits

(40%) ¤ b. Would have manufacturer conduct "special" tests to my limits

(20%) ¤ c. Would accept manufacturer's results and limits

(0%) ¤ d. Would not change current techniques

It is apparent from the response to U7 that filter users are ready to change their filter quality control techniques. This shows the industry's desire for a standard.

The final portion of the survey was devoted primarily to the filter manufacturers. The first four questions related to their present quality control procedures.

SPECIAL SECTION FOR FILTER MANUFACTURERS

M1. Are your customers currently specifying quality control limits for your elements?

Most ¤ (39%) Only a few ¤ (50%) None ¤ Comments:

M2. Do you currently use quality control testing techniques on your finished elements?

All ¤ (44%) Some ¤ (50%) None ¤ Comments:

M3. Do most of your customers accept your test data or do they conduct further testing?

Accept tests ¤ (68%) Retest ¤ (32%) No opinion ¤ Comments:

M4. What limits of maximum percent defective do you allow for your manufactured elements?

Most critical elements (0.17% defective) Least critical elements (2.2% defective)

Currently, 39% of the manufacturers responding find that most of their customers are specifying Q.C. limits for filter elements. Another 50% reported that a few of their cumstomers were specifying such limits (M1). This corresponds well with the response to Question M2; at present 44% of the manufacturers reporting use Q.C. testing techniques on all their finished filter elements with an additional 50% using such techniques on some of their elements. Manufacturers are aware that about 32% of their customers retest their filters. In the remaining 68% of the cases, the manufacturer believes that their Q.C. tests are sufficient to the point where the customer does not retest (M3). Filter manufacturers, in general, seemed to have tighter Q.C. limits with 0.17 percent defective for the most critical elements and 2.2 percent defective for the least critical elements, on the average (M4).

The response to Question M5 below, indicated that manufacturers tend to use bubble point heavily (80%) on their closely controlled filters and to a lesser extent (47%) on their less stringent filters. Other tests were, in order of preference: the multi-pass test,

differential pressure vs liquid flow, and the single pass test. Batch variations were not generally expected except in the multi-pass test.

M5. What method do you use for the evaluation of the quality of your most and least carefully controlled elements and do the results vary significantly from batch-to-batch of the same filter?

	Test Method	Most Controlled	Least Controlled	Batch-to-Batch Variations Yes	Batch-to-Batch Variations No
a.	Bubble point	☐ (80%)	☐ (47%)	☐ (0%)	☐ (47%)
b.	Boiling point	☐ (7%)	☐ (0%)	☐ (0%)	☐ (7%)
c.	ΔP vs air flow	☐ (20%)	☐ (7%)	☐ (0%)	☐ (13%)
d.	ΔP vs liquid flow	☐ (47%)	☐ (7%)	☐ (0%)	☐ (27%)
e.	Multi-pass test	☐ (53%)	☐ (13%)	☐ (13%)	☐ (20%)
	β	☐ (60%)	☐ (20%)	☐ (13%)	☐ (13%)
	Capacity	☐ (60%)	☐ (20%)	☐ (13%)	☐ (20%)
f.	Single-pass test	☐ (20%)	☐ (0%)	☐ (0%)	☐ (13%)
	Efficiency	☐ (20%)	☐ (0%)	☐ (0%)	☐ (13%)
	Capacity	☐ (13%)	☐ (0%)	☐ (0%)	☐ (13%)
g.	Other	☐	☐	☐	☐

M6. What percentage of the filter cost (your selling price) is to cover the cost of quality control inspections?

Most carefully controlled (7½% of cost)

Least carefully controlled (3.4% of cost)

The manufacturers presently consider quality control procedures to account for 7½ percent of the cost of the finished element (M6). The cost of quality control was reported as high as 30% of the selling price of some filters. Less critical elements had an average Q.C. cost of approximately 3.4% of the finished element cost.

M7. What tests for quality control do you currently use or specify for incoming filter media?

M8. How many filters do you manufacture from one roll of media?

Largest no. (3500) Average no. (1100) Smallest no. (150)

Responses to Question M7 indicated that the majority of the filter manufacturers use most, or all, of the TAPPI procedures in conjunction with such tests as bubble point, air flow, boiling point, and patch dirt capacity tests. The maximum number of filters reported as being made from one roll of filter media was 3500, the minimum 150, with an average of about 1100 (M8). Different rolls of media are sometimes considered different batches, even if used in the same production run. Manufacturers felt that filtration ratio, capacity, and bubble point changes due to various locations within a roll of media were small or negligible, provided that the media had not been damaged or distorted.

M9. Would you expect to find variations in the following element performance parameters for filters manufactured from various locations on a roll of filter media?

Parameter	Large Change	Small Change	Negligible Change
Filtration ratio	☐ (0%)	☐ (40%)	☐ (33%)
Capacity	☐ (0%)	☐ (40%)	☐ (33%)
ΔP vs flow	☐ (0%)	☐ (33%)	☐ (40%)
Bubble point	☐ (0%)	☐ (27%)	☐ (47%)
Others	☐	☐	☐

M10. How would a rejected lot of elements (e.g., 1000 elements) generally be treated if they failed a quality control check?

(50%) a. ¤ Destroy elements

(11%) b. ¤ Sell to buyer with less stringent control limits

(11%) c. ¤ Re-rate at a lower performance value

(28%) d. ¤ Other

M11. If a standard (NFPA, SAE, ANSI, ISO, etc.) procedure were developed for measuring filter element quality, would your present techniques be affected?

(7%) a. ¤ Would start using test standard method

(43%) b. ¤ Might use method, if appropriate

(43%) c. ¤ Would use method only if required by major customers

(0%) d. ¤ Would not use method

The response to Question M10 would suggest that the production and rejection of a defective lot of elements would be quite expensive to at least half of the filter manufacturers who destroy the defective elements. Indications are, from M11, that manufacturers are less anxious for a standard quality control program to be developed. Comments suggested that they were concerned about the cost and complexity of such a plan.

CHAPTER IV

INDUSTRIAL ADVISORY MEETINGS

Industrial advisory group meetings were held in Chicago on December 6 and 7, 1977. The two meetings allowed manufacturers and buyers to meet separately. The goals of the project and the first draft proposal were presented. The results of the discussions are incorporated in the following text where appropriate. The minutes of the meetings are presented in Appendix C. The organizations represented at the meetings were the following:

End-Item Advisory Group	Manufacturers Task Force Group
MERADCOM	MERADCOM
Oklahoma State University	Oklahoma State University
Ford Tractor	Schroeder Brothers Corporation
Clark Equipment Company	Parker Filter
WABCO CEMEG	Fram Corporation
John Deere Product Engineering	Purolator Corporation
John Deere, Dubuque Works	Nelson Filter Corporation
International Harvester	Pall Corporation
	Fleatguard, Inc.
	Donaldson Company, Inc.
	James River Paper Company

The first draft of the proposed quality control standard, composed after the first industrial survey, was presented at the industrial advisory group meetings. Some of the major points considered during formulation were: the test must determine the effectiveness of the filter as a particle separator, the particle separation characteristics should relate to multi-pass data, the test results should relate to field data where possible, the method should be as simple as possible and should conform to the questionnaire responses. The plan, as originally proposed, (Appendix E) involved the selection of filter element samples per MIL STD 414. The samples were to be evaluated by flowing contaminated oil through each of the filters and measuring both pressure drop and efficiency. The test as initially proposed would take approximately 10 minutes per element to perform, if the test facility were properly designed and operated.

The presentation of the proposed standard sparked some lively discussion at the end item user and manufacturer industrial advisory meetings. One of the major points of concensus was that the only proper way to evaluate a filter element is to test it's ability to remove and hold contaminant, it's primary function. This, it was agreed, should be the main consideration in any filter test method. It was determined that initial ΔP, particle capturing ability, dirt capacity and collapse pressure were vital parameters and should be

monitored on a Q.C. basis. It was stated by one individual that collapse pressure was in some instances, a more important parameter than efficiency because of the potential cost of failures. Discussion revealed that the sample sizes used for destructive filter testing in industry were often very much smaller than those required by MIL STD 414. It was proposed that another sampling plan be used, requiring smaller sample sizes, and/or that the sampling plan be deleted from non-military versions of the plan. The first draft standard required only one point to be evaluated for efficiency. It was proposed that more information could be gained if particle analysis was conducted at two or more points during the "life" of the element. An evaluation of efficiency after a slight differential pressure increase would provide information with respect to the filter's performance in this range. Other evaluations at a higher differential pressure were deemed necessary by most attendees. Such a test could easily be performed in conjunction with a collapse-burst evaluation. It was proposed that an upstream gravimetric level of 100 mg/ℓ or less might be valid for loading purposes. The test system, as originally proposed was felt by most to require extensive evaluation and verification. It was observed that the proposed standard, with modifications resulting from the discussion, was evolving into a modified multi-pass test. Therefore, other considerations involved in the desirability of the multi-pass test for Q.C. purposes were its wide acceptance and use

throughout industry, the fact that the test concepts have been verified repeatedly, and the existance of a substantial number of multi-pass test facilities. It was concluded that a "standard" multi-pass test should be the basis for the Q.C. test. Modifications to accelerate the multi-pass test were suggested to achieve substantial time savings; however, most modifications require verification. Further discussion revealed that industry feels a need for a standardized method of in-line particle counting and that the development of a standard in this area would be a significant contribution.

CHAPTER V

INDUSTRIAL SURVEY II, FORMULATION AND RESPONSE

Most manufacturers and buyers attending the advisory meetings were not in a position to discuss their current sampling procedures in detail. It was concluded that another survey was in order, and that the primary thrust of the survey should be to determine the common sample frequencies and sample sizes for various lots or batches and also the source and availability of their Q.C. plan. This survey is presented as Appendix D.

The questionnaire was sent to 80 individuals representing over 75 different corporations and/or divisions. Thirty-one responses were received, by return of the questionnaire and by telephone. The degree to which the desired information was given varied greatly between respondants. Some provided most of the information requested. Others, often responding by telephone, informed us that their only Q.C. procedure other than visual inspection at installation, was an "occasional" multi-pass test. Of those providing sufficient information for analysis, the manufacturers were observed to have a markedly different philosophy on the subject of quality control. In light of this difference, the manufacturers and users responses are discussed separately, where appropriate. Each of the summaries is based upon the information provided in response to a specific question and may be the result of

as few as three opinions. If less than three responses were received, no summary was made.

The first part of the survey was an introductary paragraph and request for general information, identical to that of the previous survey. The second part of the questionnaire consisted entirely of questions related to present quality control plans and was designed for manufacturers and users alike.

The first questionnaire primarily surveyed opinions related to Q.C. procedures so it was deemed necessary to determine specifics in relation to current Q.C. procedures. The first question was used to determine the present degree to which Q.C. plans, in general, are used. It was reported that 100% of the manufacturers are using some form of quality control procedures, while only about one-fourth of the users currently use Q.C. plans consistently.

1. Is your company/division consistently using a quality control plan(s) on hydraulic filter elements (fluid-power type)?

   Yes (100% manufacturers) (24% users) No

2. Does your Q.C. plan(s) monitor:

   Variables (Rating items on a continuous measurement scale)

   Attributes (64% manufacturers) (Rating items on a good/bad basis)

The response to Question 2 revealed that the majority of the manufacturers (64%) monitored attributes in preference to variables. User response was limited, such that no generalizations could be made.

3. If you use a "nondestructive" Q.C. test on hydraulic, fluid-power filter elements, how many samples would be inspected and what average quality limit would be utilized for the following lot sizes under your present plan(s)?

   Lot of 50 Sample Size____pcs. A.Q.L.%_____

   Lot of 200 Sample Size____pcs. A.Q.L.%_____

   Lot of 750 Sample Size____pcs. A.Q.L.%_____

   Lot of 1500 Sample Size____pcs. A.Q.L.%_____

   Lot of 5000 Sample Size____pcs. A.Q.L.%_____

   Comments:

4. What "nondestructive" test(s) is used?

The response to question three indicated that manufacturers generally use a plan similar to MIL STD 105-D to determine sample size for nondestructive testing while buyers sample much less frequently. A.Q.L's reported ranged from 1.5% to 4%.

The nondestructive test used by manufacturers in almost every case was bubble point. Dimensions were also inspected by the above plans.

5. If you use a "destructive" Q.C. test on hydraulic, fluid-power filter elements, how many samples would be inspected and what average quality limit would be utilized for the following lot sizes under your present plan(s)?

Lot of 50	Sample Size_____pcs.	A.Q.L%_____
Lot of 200	Sample Size_____pcs.	A.Q.L%_____
Lot of 750	Sample Size_____pcs.	A.Q.L%_____
Lot of 1500	Sample Size_____pcs.	A.Q.L%_____
Lot of 5000	Sample Size_____pcs.	A.Q.L%_____

Comments:

6. What "destructive" test(s) is used?

It was found that manufacturers and buyers, under their present systems, use destructive tests infrequently. The tests that run are generally considered "for verification only" and no A.Q.L's are generally specified. The destructive tests commonly used are: standard multi-pass, flow fatigue, material compatibility and collapse/capacity tests.

7. Is the quality measure of an individual, hydraulic, fluid-power, filter element an event that is independent of the quality measure of other filters in the same batch?

Yes No (100% manufacturers) Comments:

A vote of 100% "no" in response to Question 7 indicates that all manufacturers responding feel that the quality of a specific filter, in a batch, is closely related to the quality of other filters within the batch. The assumption that all the filters within a batch are closely related seems to be common. Such an assumption could lead to a drastic reduction in the number of samples necessary for any confidence level. It is believed that it is this assumption that allows manufacturers to perform destructive tests infrequently (see Question 5 and associated discussion) while maintaining high confidence in product quality.

8. If your Q.C. plan(s) is published or "standard," where could it be found?
9. If your Q.C. plan(s) was developed specifically for your use, please describe the rationale behind the plan(s) or, if possible, enclose a copy of the plan(s).
10. Briefly, state your reasons for choosing the Q.C. plan that you utilize.

So few responses to Questions 8, 9, and 10 were received that no summary or generalizations could be made with any statistical validity.

The majority of the Q.C. plans presently used are confidential, "in house" plans. A few Q.C. plans were sent to the FPRC by those responding; however, the majority of these plans were devoted to

monitoring exterior qualities of the finished element, e.g., dimensions, finish, markings and packaging. In the plans received, little direct attention was paid to monitoring the ability of the unit to perform it's vital function, the removal of contaminants from the system.

The majority of those responding to Question 10 felt that since their warranty costs were low, they could not justify a more expensive plan. Some manufacturers felt that consumer risk was small and had adjusted their Q.C. plans accordingly.

11. Briefly, describe your present "in process" Q.C. inspections made during element fabrication.

12. To what degree do the "in process" inspections contribute to end-product quality?

It was apparent, from the response to Questions 11 and 12, that manufacturers feel that "in process" inspections help them produce better filters at lower costs. A manufacturer may have well over 50 different inspections and checks made before a batch of elements is completed. It is felt "in process" inspections are essential to the production of a quality Filter Element.

CHAPTER VI

SECOND DRAFT PROPOSAL

Following the industrial advisory group meetings a second draft proposal was formulated. The opinions expressed at the industrial meetings were those of the companies most concerned about the test method. It is likely that these companies would be among the first to use the proposed standard. For this reason, the second draft test method was tailored to fit the needs expressed at the meetings.

It was agreed that the Quality Control Test Method should resemble the ISO, ANSI, NFPA Multi-pass Test as nearly as possible. Yet the standard multi-pass test was considered too time consuming in many cases, and the Quality Control Test method should have provisions for the acceleration of the standard tests. A revision of the standard multi-pass test entitled "Accelerated Hydraulic Filter Element Multi-pass Test For Quality Control" is presented as Appendix F. The following comments are a discussion of the changes made in or deviations from the standard procedure. The section(s) discussed are referenced by clause number for easy reference to Appendix F.

1.1 The scope of the procedure is changed to include "an accelerated" test and "for quality control."

2. The purpose is modified to include the changes in scope.

8.4. Inline particle counting, conducted in a yet to be standardized manner, would be a viable alternative to the use of sample bottles and associated analysis techniques.

11.1. The clause involving the bubble point test in the multi-pass standard (11.1.2) is not felt to be required for the proposed multi-pass test procedure for Q.C. purposes.

11.2.1 The upstream gravimetric level will be determined by the Filter Q.C. specification and shall be between 3 and 50 milligrams per liter. Variable upstream gravimetric levels will allow test duration to be optimized. The level specified for evaluation of a filter to be evaluated according to this procedure will remain constant for purposes of comparison.

A note is included discussing the variable injection rate and its relationship to test duration. Past studies with lubricating oil filters[1] show that variations in base upstream gravimetric level within the limits above, have little effect on the ACFTD capacity of a filter element, but show great reductions in test duration.

11.2.3. The method of determining the necessary injection gravimetric level is modified to incorporate the variable base upstream gravimetric levels.

11.3.4. It is felt, that in order to decrease clean-up time for filters whose specified β_{10} is less than 2, a background contamination level of less than 150 particles per millilitre greater than 10 micrometres is acceptable.

11.3.5. Sample flows for bottle samples may differ from those necessary for inline particle analysis. Inline particle counters should be installed according to the applicable standard or specification.

11.3.6. Sample flow from upstream of the filter is to be returned to the reservoir when bottle samples are not being taken.

12.4. Initial cleanliness requirements may be verified by bottle sample or inline analysis.

12.12. For those wishing to evaluate the collapse-burst parameters, the following sentence was included. "Continue injection to collapse or maximum pressure specified. This step is optional, refer to specification."

12.14. If filter tests are run consecutively, the number of injection gravimetric analysis may be decreased by taking an injection sample before each filter is tested and after a test in the following cases: if the filter just tested is the final filter of the series, if the system will be

shut down before testing continues, or if more than 1 hour will elapse before the next filter is tested.

14.1. The standard multi-pass test calls for particle counts at the 10, 20, 30 and 40 micrometre levels. For many filters, analysis of other sizes will provide valuable information. The sizes to be analyzed should be set forth in the specification.

14.2.1. Before the start of each test, an injection sample is taken. If an individual filter test is followed by another, a sample will be taken at that point; if it is not to be followed by such a test, a sample is to be taken per 12.14. This yields two injection gravimetric levels for each filter analysis with a minimum number of samples. These two levels may be averaged to determine average injection gravimetric level.

14.4.1. Base upstream gravimetric levels for test are acceptable if within ± 10% of the specification value.

CHAPTER VII

TEST RESULTS & DISCUSSION

The testing program was limited in both scope and extent. The tests performed were designed to give an overview of the parameters tested and suggest trends that might be confirmed by further research at a later date. Because of the similarity of the Q.C. test to the standard multi-pass test method which has been verified and is accepted throughout industry, it was felt that the proposed Q.C. test would require reduced verification effort. Filter element donations for testing purposes were solicited at the industrial meetings. This resulted in three manufacturers donating elements for test purposes.

One of the more radical changes in filter evaluation procedure proposed was the elimination of the bubble point test for fabrication integrity. This test is part of the internationally accepted standard multi-pass test and is a popular test among filter users and manufacturers alike, presumeably because of its simplicity. Many of the groups surveyed felt that the test was good for fabrication integrity alone and that there was no correlation between the bubble point and the vital parameters of efficiency, capacity and pressure drop. Four filter elements from one manufacturer with varying bubble points, but apparently indentical in every other way, were tested. A summary of the results of these tests are shown in

Fig. 7-1 and the complete multi-pass test data is presented in Appendix G for all tests conducted. No significant relationship is apparent to relate bubble point to particle separation efficiency. If there is truly no relationship between bubble point and efficiency then the test is of little use except to indicate major element damage or poor fabrication. The bubble point test has never been accepted as an indicator of capacity or pressure drop.

In order to utilize statistical sampling plans such as MIL-STD-414, the parameter evaluated is required to be normally distributed. A number of filters were tested to evaluate the normality of various parameters within a given group of elements. The relief valve within the element housing was sealed to eliminate leakage. When the first element of group #1 was tested, its pressure increased to approximately 29% of the terminal value and stabilized. The test was terminated at this point. The element, when inspected, showed signs of leakage around the end cap seals. This filter when compared to other filters of the same type was found to be noticeably shorter. The remaining tests on group #1 involved the use of additional spacers and gaskets to help insure good end cap sealing. The results of these tests indicate that leakage is occurring, despite efforts to prevent it. Leakage is suspected whenever a filter exhibits a relatively constant beta with increasing particle size as shown in the data for element group #1 in Appendix G.

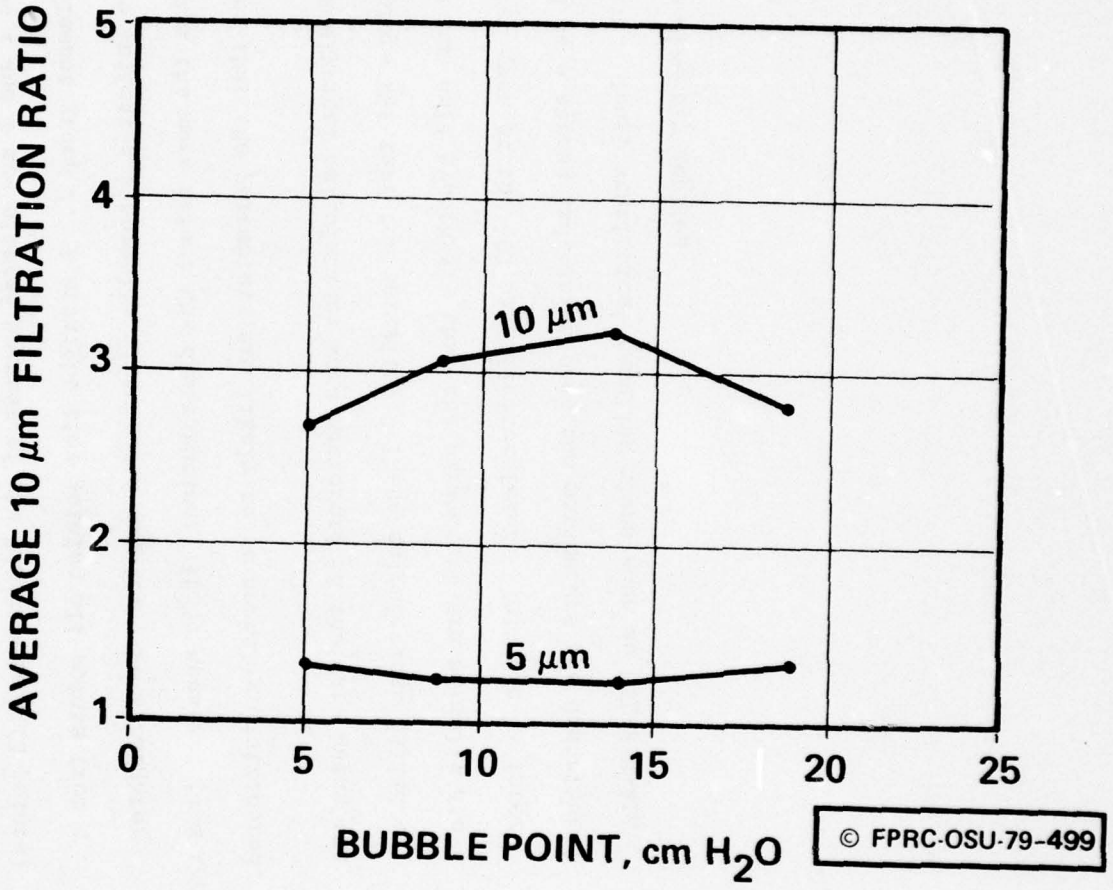


Fig. 7-1. Effect of Bubble Point on Filtration Ratios.

A total of fifteen filters from three different manufacturers were tested. The data was analyzed to determine the normality of the distribution. Figures 7-2 and 7-3 present filtration ratio and capacity data plotted on normal probability paper. Only element groups 1 and 3 are plotted because of the limited data (3 points) for element group 2. A straight line joining all points from a sample indicates a normally distributed sample. The straighter the line the more normal the distribution. It is apparent from the figures that the parameters are likely to be normally distributed.

Analytical calculations were performed on the data using the Lilliefors K-F test for normality.[2] These calculations failed to reject the null hypothesis that the parameters are normally distributed even at the 10% significance level. This can be interpreted as a strong indication that the parameters are distributed normally. Thus, statistical sampling plans such as MIL-STD-414 can be directly applied.

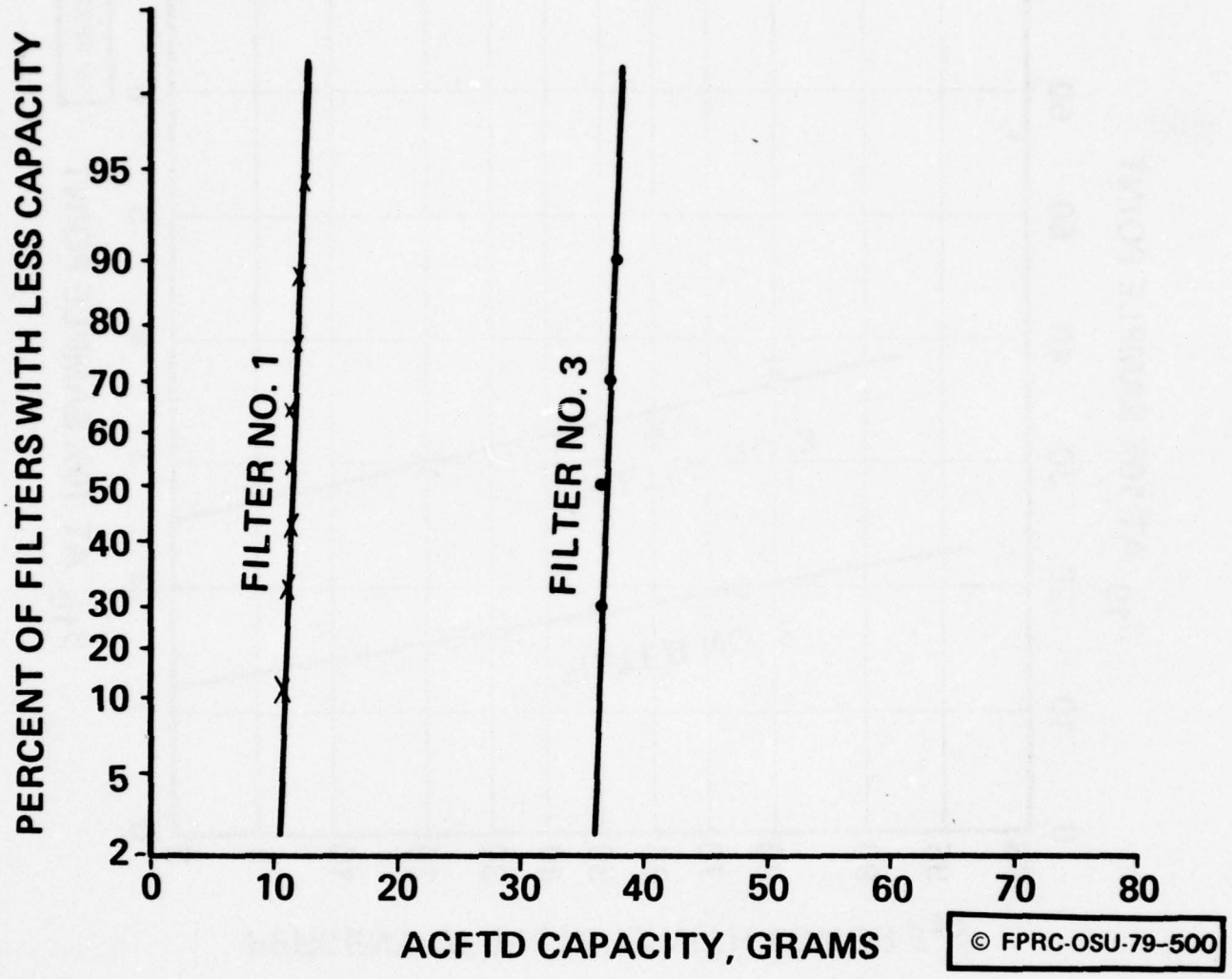


Fig. 7-2. Normal Probability Plot of Filtration Ratios

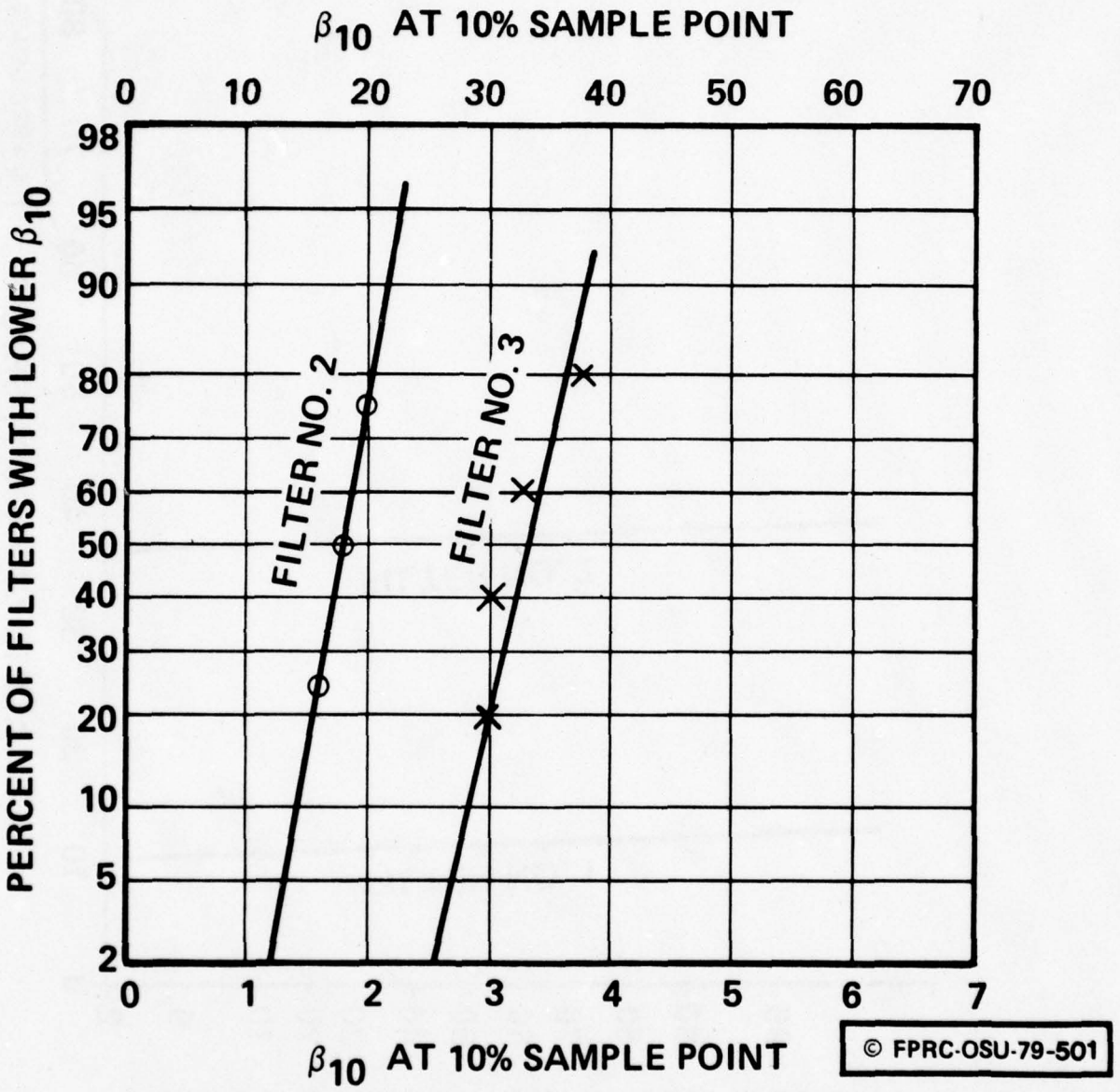


Fig. 7-3. Normal Probability Plot of ACFTD Capacities.

CHAPTER VIII

CONCLUSIONS AND RECOMMENDATIONS

The primary objective of the MERADCOM-OSU Hydraulic Filter Quality Control Program was to develop a test procedure, applicable to quality control of hydraulic filters, which was industrially acceptable and compatible with the U.S. Army MERADCOM requirements. It is felt that this objective has been successfully fulfilled. A procedure has been developed for inclusion in a hydraulic filter specification. The basis of this performance test — the accelerated multi-pass particle separation test — was developed on the foundation established by the multi-pass test for hydraulic filters, which was a product of an earlier MERADCOM-OSU Program. A limited testing program was conducted in order to explore the various test parameter values for the accelerated multi-pass method for hydraulic filter evaluation.

The filter quality control project was originally designed as a two-part effort. The first part of this effort, the development of a test method, acceptable to industry, for monitoring the conformance of filter elements to specifications on a Q.C. basis, has been completed, and is the subject of this report. The second part of the program was to involve an extensive test and evaluation effort in order to refine and affirm the concepts developed during the first part of the project. It is felt that because of the test

similarity to the multi-pass test method, this verification effort can be significantly reduced.

Industry, especially filter users, have expressed their desire for a filter quality control standard, like the one developed during the project. It is expected that further testing and analysis will be necessary before this method will be widely accepted for quality control throughout industry. Acceptance of this standard by industry would allow the government to procure filters, manufactured in accordance with specifications, without the use of a qualified products list.

Initial attempts have already been made by OSU personnel to promulgate these procedures, ideas, and results on an industrial level in technical circles. The end-item users and manufacturers were informed of the program through the industrial advisory meetings. Members of the OSU staff have been present at SAE and NFPA committee meetings and are actively participating in the ISO subcommittee dealing with filter test methods. Hopefully, after the procedure is fully verified, the filter industry will be motivated to pursue the acceptance of these methods as national and international standards.

It is recommended that, in order for MERADCOM to receive full benefit from this initial test development effort, a continued

effort should be expended to enhance the probability of adoption at the national and international levels. Additional verification tests should be performed with the abbreviated multi-pass test on a broad range of hydraulic filters. The repeatability and reproducibility of the test method should also be established.

SELECTED REFERENCES

1. *"Lubricating Oil Filtration Study for Mobile On-Off Highway Diesel Engine-Driven Vehicles,"* Annual Report to U.S. Army MERADCOM, Contract No. DAAK02-72-C-0172, Oklahoma State University, December, 1974.

2. Locks, M. O., Reliability, Maintainability and Availability Assessment, Hayden Books Company, Inc., Rochelle Park, New Jersey, 1973.

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APPENDIX A

GENERAL BIBLIOGRAPHY

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1. Fochtmon, Edward G. and Victor R. Iranuski, *"Nondestructive Test of Cartridge Filter Elements,"* Filtration & Separation, September/October 1971, pp. 608-610.

2. King, R. K., R. K. Tessmann, and G. A. Roberts, *"A Look at Quality Control Procedures for Filters,"* Fluid Power Research Conference, Oklahoma State University, Paper P76-17.

3. Fochtmon, Edward G., *"Cartridge Filtration: Technology and Economics,"* Filtration and Separation, May/June 1973, pp. 289-294.

4. Pall, Dr. David B., *"Quality Control of Absolute Bacteria Removal Filters,"* Pall Bulletin AB700-12-73, Pall Corporation.

5. Horrocks, A. C., and D. E. Corner, *"Contamination Control and Measurement in Aerospace Hydraulic Systems,"* Filtration and Separation, March/April 1973, pp. 193-199, 224.

6. Schmitz, James E., *"Particle Size Analysis of Oil Samples with a Coulter Counter,"* Filtration and Separation, July/August 1973, pp. 426-428.

7. Harcourt, R. A. G., *"Development and Testing of a High Efficiency Hydraulic Filter Element,"* Filtration and Separation, January/February 1973, pp. 67-73.

8. Haulsein, Robert H., *"Verification of the Reliability of Filter Cartridge Assemblies,"* Filtration and Separation, March/April 1976, pp. 161-163.

APPENDIX B

INDUSTRIAL QUESTIONNAIRE I

PLEASE RETURN BY 15 NOVEMBER 1977

INDUSTRIAL QUESTIONNAIRE

Hydraulic Filter Element Quality Control Evaluation

This questionnaire is being conducted as a part of the U.S. Army MERADCOM/Oklahoma State University project to develop quality control evaluation methods for hydraulic filter elements. Please complete the appropriate sections of this questionnaire and return to:

Mr. Eric Eskilson, Project Leader – Filter Q. C.
Fluid Power Research Center
Oklahoma State University
Stillwater, Oklahoma 74074
(405) 624-7375

Name of Respondent: _____ Title: _____

Company/Division: _____ Phone: _____

Address: _____

City: _____ State: _____ Zip Code: _____

I do ☐ I do not ☐ wish to participate in future industrial advisory meetings.

My company/division should be considered as a manufacturer ☐ end-item user ☐.

GENERAL SECTION FOR BOTH USERS AND MANUFACTURERS

G1. How many types (different part numbers) of hydraulic filters does your company deal with (manufacture or receive) on a typical business day?

Minimum no. _____ Average no. _____ Maximum no. _____

Comments: _____

G2. Would the manufacture or receipt of any given type of filter element normally be considered a batch or continuous process?

Batch ☐ Continuous ☐ No opinion ☐ Comments: _____

G3. How many types of hydraulic filters do you produce or buy only on a speciality basis?

_____ Comments: _____

G4. How many types of hydraulic filters do you produce or buy on a regular or continuous basis?

_____ Comments: _____

G5. What is, typically, the number of filter elements of a given type manufactured or purchased at one time?

Highest production _____ Average production _____ Lowest production _____

Comments: _____

G6. Do your quality control requirements vary from one element type to another?

Yes ☐ No ☐ No opinion ☐ Comments: _____

G7. What type of method do you use to refer to a filter element's particle separation characteristics when buying or selling?

Filtration ratio (β) ☐ Absolute rating ☐
Nominal rating ☐ Mean pore size ☐
Efficiency (how measured) _____

Other (describe) _____

G8. Do you feel that a valid filter quality control test method should produce results which can be directly correlated with particle separation characteristics?

Yes ☐ No ☐ No opinion ☐ Comments: _____

G9. Please check the blanks below if you feel that the following non-destructive tests are valid indicators of fabrication integrity and/or particle separation characteristics. If a test is applicable to a given special type of filter, please specify that type.

Test Method	Fabrication Integrity		Separation Characteristics		Comments
	All Filter Types	Special Types	All Filter Types	Special Types	(List Special Types)
Bubble Point	☐	☐	☐	☐	_____
Boiling Point	☐	☐	☐	☐	_____
Air Permeability (ΔP vs flow)	☐	☐	☐	☐	_____
Liquid Permeability (ΔP vs flow)	☐	☐	☐	☐	_____
Tracer Gas (ammonia/sock)	☐	☐	☐	☐	_____
Others					
_____	☐	☐	☐	☐	_____
_____	☐	☐	☐	☐	_____

G10. Do you feel that the dirt capacity of a filter is an important parameter to be monitored in a quality control check?

Yes ☐ No ☐ No opinion ☐ Comments: _____

G11. Do you feel that minor capacity variations can be noted by monitoring the number of pleats in a pleated cartridge?

Yes ☐ No ☐ No opinion ☐ Comments: _____

G12. Is clean element ΔP (in a test liquid) an important parameter in a quality control check?

Yes ☐ No ☐ No opinion ☐ Comments: _____

G13. What is your opinion of the maximum allowable time for a quality control test on a filter element?

< 1 minute ☐ 1 - 5 minutes ☐ 5-10 minutes ☐ 10 - 15 minutes ☐ >15 minutes ☐ Other ____

Comments: _____

G14. If a procedure could be devised that would expose a filter to a contaminant for separation appraisal then remove all the contaminants, could the filter element be considered "new" and be sold or purchased as such?

Yes ☐ No ☐ No opinion ☐ Comments: _____

SPECIAL SECTION FOR FILTER USERS AND BUYERS

U1. Do you currently find filter manufacturers specifying quality control limits for the elements they are marketing?

Most ☐ Only a few ☐ None ☐ Comments: _____

U2. Do you currently specify given limits of the quality of incoming elements?

All ☐ Some ☐ None ☐ Comments: _____

U3. What limits of maximum percent defective do you allow for purchased elements?

Most critical elements _____ % Least critical elements _____ % Comments: _____

_____

U4. Do you accept manufacturers'data for establishing Q. C. or do you plan to conduct your own tests?

Manufacturers' tests ☐ Conduct tests ☐ Plan to test ☐ Comments: _____

_____

U5. What method do you use or specify for the evaluation of the quality of purchased elements for your most and least critical applications?

Test Method	Most Critical Application	Least Critical Application	Comments
a. Bubble point	☐	☐	
b. Boiling point	☐	☐	
c. ΔP vs air flow	☐	☐	
d. ΔP vs liquid flow	☐	☐	
e. Multi-pass test	☐	☐	
β	☐	☐	
Capacity	☐	☐	
f. Single-pass test	☐	☐	
Efficiency	☐	☐	
Capacity	☐	☐	
g. Other (specify)			
_____	☐	☐	

Comments: _____

U6. What are your typical costs for quality control testing expressed as a percentage of total element cost?

Most critical application _____% Least critical application _____%

Comments: _____

U7. If a standard (NFPA, SAE, ANSI, ISO, etc.) procedure were developed for measuring filter element quality control, how would your purchase and acceptance techniques be affected?

a. ☐ Would start conducting standard tests and specifying limits
b. ☐ Would have manufcturer conduct "special" tests to my limits
c. ☐ Would accept manufacturers' results and limits
d. ☐ Would not change current techniques

Comments: _____

SPECIAL SECTION FOR FILTER MANUFACTURERS

M1. Are your customers currently specifying quality control limits for your elements?

Most ☐ Only a few ☐ None ☐ Comments: _____

M2. Do you currently use quality control testing techniques on your finished elements?

All ☐ Some ☐ None ☐ Comments: _____

M3. Do most of your customers accept your test data or do they conduct further testing?

Accept tests ☐ Retest ☐ No opinion ☐ Comments: _____

M4. What limits of maximum percent defective do you allow for your manufactured elements?

Most critical elements _____ % Least critical elements _____ % Comments: _____

_____

M5. What method do you use for the evaluation of the quality of your most and least carefully controlled elements and do the results vary significantly from bath-to-batch of the same filter?

Test Method	Most Controlled	Least Controlled	Batch-to-Batch Variations Yes	Batch-to-Batch Variations No
a. Bubble point	☐	☐	☐	☐
b. Boiling point	☐	☐	☐	☐
c. ΔP vs air flow	☐	☐	☐	☐
d. ΔP vs liquid flow	☐	☐	☐	☐
e. Multi-pass test	☐	☐	☐	☐
β	☐	☐	☐	☐
Capacity	☐	☐	☐	☐
f. Single-pass test	☐	☐	☐	☐
Efficiency	☐	☐	☐	☐
Capacity	☐	☐	☐	☐
g. Other (specify) _____	☐	☐	☐	☐

Comments: _____

M6. What percentage of the filter cost (your selling price) is to cover the cost of quality control inspections?

Most carefully controlled _____ % Least carefully controlled _____ %

Comments: _____

M7. What tests for quality control do you currently use or specify for incoming filter media?

_____

_____

M8. How many filters do you manufacture from one roll of media?

Largest no. _____ Average no. _____ Smallest no. _____

Comments: _____

M9. Would you expect to find variations in the following element performance parameters for filters manufactured from various locations on a roll of filter media?

Parameter	Large Change	Small Change	Negligible Change
Filtration ratio	☐	☐	☐
Capacity	☐	☐	☐
ΔP vs flow	☐	☐	☐
Bubble point	☐	☐	☐
Others _____	☐	☐	☐
_____	☐	☐	☐

Comments: _____

M10. How would a rejected lot of elements (e.g., 1000 elements) generally be treated if they failed a quality control check?

a. ☐ Destroy elements
b. ☐ Sell to buyer with less stringent control limits
c. ☐ Re-rate at a lower performance value
d. ☐ Other _____

Comments: _____

M11. If a standard (NFPA, SAE, ANSI, ISO, etc.) procedure were developed for measuring filter element quality, would your present techniques be affected?

a. ☐ Would start using test standard method
b. ☐ Might use method, if appropriate
c. ☐ Would use method only if required by major customers
d. ☐ Would not use method

Comments: _____

APPENDIX C

COMBINED MINUTES FROM HYDRAULIC FILTER ELEMENT QUALITY CONTROL INDUSTRIAL ADVISORY MEETINGS

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COMBINED MINUTES FROM HYDRAULIC FILTER ELEMENT

QUALITY CONTROL INDUSTRIAL ADVISORY MEETINGS

Dec. 6 End Item Manufacturers/Users Meeting
Dec. 7 Filter and Media Manufacturers

1. Both meetings were called to order at 9:45.
2. Attached are attendance lists.
3. Opening remarks were made by Dr. Leonard Bensch, OSU, FPRC and MERADCOM. Their primary focus was the army's need for a method, acceptable to industry, that would assure them that filters received, in quantity, were consistantly built to the specifications required. They stated that their goal was to produce a standard that would be used by buyers and sellers throughout industry, and allow the army to be assured of conformance to specifications when purchasing under the same plan.
4. A summary of the Industrial Survey on Filter Quality Control was presented and briefly discussed.
5. Draft No. 1 of the proposed standard was presented.
6. Discussion with end item manufacturers/user on December 6 yielded the following:
   A. The only proper way to evaluate a filter is to test its ability to remove contaminant, its primary purpose.
   B. Initial ΔP, particle capturing abilities, dirt capacity, and collapse pressure should be constantly monitored.
   C. Mil 414 requires larger sample sizes than commonly used and does not account for past history of a vendor. Test method should be divorced from sampling plan(s), at least for industrial user/manufacturers.

D. More information can be gained by partially loading a filter (say to 40-80% ΔP increase), then making particle counts. It was proposed that an upstream gravimetric of 100 mg/l or less was valid for loading purposes.

7. Discussion with manufacturers on December 7 yielded the following:

   A. Sole purpose of a filter element is to remove and hold contaminant, and element quality control tests should directly reflect its ability to do so.

   B. Test system as proposed would require years to completely evaluate and verify.

   C. Test system, with modifications, as proposed by users was approaching a multi-pass test system.

   D. Rather than introduce a new test system, work should be done to shorten time necessary for multi-pass tests and evaluations.

   E. A standard should be developed covering on-line particle counting and dilution.

   F. It was agreed that test procedures should not include sampling plan(s) except in military version(s).

   G. A survey should be conducted to determine sampling plan(s) available and how they are currently used with regard to hydraulic filter element quality control.

8. Next meeting(s) should include both manufacturers and users and would be held in Chicago during *first quarter* of 1978.

APPENDIX D

INDUSTRIAL QUESTIONNAIRE I

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PLEASE RETURN BY 28 February 1978

INDUSTRIAL QUESTIONNAIRE

Hydraulic Filter Element Quality Control Evaluation

This survey of industrial practice is being conducted as a part of the U.S. Army MERADCOM/ Oklahoma State University project to develop quality control methods for hydraulic filter elements. Please complete the questionnaire and return to:

Mr. Eric Eskilson, Project Leader – Filter Q. C.
Fluid Power Research Center
Oklahoma State University
Stillwater, Oklahoma 74074
(405) 624-7375

Name of Respondent: _____ Title: _____

Company/Division: _____ Phone: _____

Address: _____

City: _____ State: _____ Zip Code: _____

My company/division should be considered as a manufacturer ☐ end-item user ☐.

1. Is your company/division consistently using a quality control plan(s) on hydraulic filter elements (fluid-power type)?

   Yes ☐ No ☐ Comments: _____

2. Does your Q. C. plan(s) monitor:

   Variables ☐ (Rating items on a continuous measurement scale)
   Attributes ☐ (Rating items on a good/bad basis)

   Comments: _____

3. If you use a "nondestructive" Q. C. test on hydraulic, fluid-power filter elements, how many samples would be inspected and what average quality limit would be utilized for the following lot sizes under your present plan(s)?

					Comments:
Lot of 50	Sample Size	_____ pcs.	A.Q.L.%	_____	
Lot of 200	Sample Size	_____ pcs.	A.Q.L.%	_____	_____
Lot of 750	Sample Size	_____ pcs.	A.Q.L.%	_____	
Lot of 1500	Sample Size	_____ pcs.	A.Q.L.%	_____	_____
Lot of 5000	Sample Size	_____ pcs.	A.Q.L.%	_____	

4. What "nondestructive" test(s) is used? _____

5. If you use a "destructive" Q. C. test on hydraulic, fluid-power filter elements, how many samples would be inspected and what average quality limit would be utilized for the following lot sizes under your present plan(s)?

Lot of 50	Sample Size	_____	pcs.	A.Q.L.%	_____
Lot of 200	Sample Size	_____	pcs.	A.Q.L.%	_____
Lot of 750	Sample Size	_____	pcs.	A.Q.L.%	_____
Lot of 1500	Sample Size	_____	pcs.	A.Q.L.%	_____
Lot of 5000	Sample Size	_____	pcs.	A.Q.L.%	_____

Comments: _____

6. What "destructive" test(s) is used? _____ _____

7. Is the quality measure of an individual hydraulic, fluid-power filter element an event that is independent of the quality measure of other filters in the same batch?

Yes ☐ No ☐ Comments: _____

8. If your Q. C. plan(s) is published or "standard," where could it be found?

_____

9. If your Q. C. plan(s) was developed specifically for your use, please describe the rationale behind the plan(s) or, if possible, enclose a copy of the plan(s).

_____

_____

_____

10. Briefly, state your reasons for choosing the Q. C. plan that you utilize.

_____

_____

_____

The following questions pertain to hydraulic (fluid power) filter manufacturers only.

11. Briefly, describe your present "in process" Q. C. inspections made during element fabrication.

_____

_____

_____

12. To what degree do the "in process" inspections contribute to end-product quality?

10 February 1978

APPENDIX E

DRAFT I, PROPOSED TEST METHOD FOR QUALITY CONTROL OF FINE HYDRAULIC FILTER ELEMENTS

REFERENCES

1. American National Standard Glossary of Terms for Fluid Power, ANSI/B93.2-1971, and Supplements thereto. [ISO/TC 131/SC 1 (USA-2) 3].
2. SI units and recommendations for the use of their multiples and of certain other units, ISO 1000-1973.
3. International Standard Graphical Symbols for Hydraulic and Pneumatic Equipment and Accessories for Fluid Power Transmission, ISO/R 1219-1970. Agrees with ANSI/Y32.10-1967.
4. American National Standard Drafting Practices for Fluid Power Diagrams, ANSI/Y14.17-1966.
5. International Standards Organization Standard Draft Proposal Method for Calibration of Liquid Automatic Particle Counters Using "AC" Fine Test Dust, ISO/TC 131/SC 6 (USA-9) 12.
6. American National Standard Procedure for Qualifying and Controlling Cleaning Methods for Hydraulic Fluid Power Fluid Sample Containers, ANSI/B93.20-1972. [ISO/TC 131/SC 6 (USA-8) 9].
7. Society of Automotive Engineers Determination of Hydraulic Pressure Drop, SAE/ARP 24B-1968.
8. American National Standard Method for Extracting Fluid Samples from the Lines of an Operating Hydraulic Fluid Power System (for Particulate Contamination Analysis), ANSI/B93.19-1972. [ISO/TC 131/SC 6 (USA-2) 3]
9. Society of Automotive Engineers Procedure for the Determination of Particulate Contamination in Hydraulic Fluids by the Control Gravimetric Procedure, SAE/ARP 785-1963.

OSU-FQC
DRAFT NO. 1
30 NOVEMBER 1977

PROPOSED TEST METHOD FOR ASSESSING THE QUALITY CONTROL OF FINE HYDRAULIC FILTER ELEMENTS

NOTE:

This test procedure is being prepared by the Fluid Power Research Center at Oklahoma State University under the sponsorship of the U.S. Army MERADCOM. Industrial comments about the method are solicited.

PROPOSED TEST METHOD FOR ASSESSING THE QUALITY CONTROL OF FINE HYDRAULIC FILTER ELEMENTS

INTRODUCTION

The service life and reliability of hydraulic systems are extremely vital to fluid power equipment users; thus, it is important to consider every aspect of design, procurement, and operation which can influence such factors. The life and reliability of system components are highly dependent upon the contamination level maintained in the system fluid. Since the system filter is the primary source of control of the contamination level, its performance has a direct influence upon the longevity of the other system components. Extensive work has been conducted on the development of an industrially accepted filter performance evaluation method in order to allow a rational selection of a specific filter for protection of a given hydraulic system.

The multi-pass test has now achieved acceptance by national and international standards bodies and has been used with great success as a performance assessment tool. Specification tests for U.S. Army and industrial qualified products lists have been based directly upon the multi-pass procedure. However, one of the primary problems with the QPL procurement method is readily apparent with hydraulic filters – the buyer does not have a convenient method for insuring that the performance of the filters being received meets the specification levels of the QPL on a continuing basis. Filter elements are easily damaged in shipment; and, unless stringent manufacturing checks are made, leaks around and through medium seals can occur.

Industry inputs and experience at the Fluid Power Research Center indicate that an overall emphasis by the fluid power industry on filter quality control is lacking and that a standardized method of readily assessing filter quality would be a great asset. Previous quality control methods (such as the bubble point, air flow, and tracer gas techniques) have not been shown to relate to actual filter performance. The following procedure, therefore, has been developed in order to evaluate filter quality control in terms compatible with actual separation performance.

1. SCOPE

1.1 To include an abbreviated single-pass filtration performance test for fine hydraulic fluid power filter elements.

1.2 To include a procedure for statistically monitoring the quality level of a group of filters with regard to particle removal characteristics and initial differential pressure.

1.3 To include a test currently applicable to hydraulic fluid power filter elements which exhibit a 10 micrometre filtration ratio less than 75.0.

2. PURPOSE

To provide a test procedure yielding statistical information usable for the determination of whether or not a group of fine hydraulic fluid power filter elements is in conformance with a predetermined specification.

3. TERMS AND DEFINITIONS

See American National Standard Glossary of Terms for Fluid Power, ANSI/ B93.2-1971, Reference No. 1.

4. UNITS

The International System of Units (SI) is used in accordance with Reference No. 2.

5. LETTER SYMBOLS

(Letter symbols are in accordance with Reference No. 2.)

6. GRAPHIC SYMBOLS

Graphic symbols used are in accordance with Reference No. 3 and Reference No. 4. Where Ref. No. 3 and Ref. No. 4 are not in agreement, Reference No. 3 governs.

7. GENERAL PROCEDURE

7.1 Set up and maintain apparatus per Sections

7.2 Run all tests per Sections

7.3 Analyze data from Sections per Sections

7.4 Present data as per Section

8. TEST EQUIPMENT

8.1 Use suitable timer for measuring minutes and seconds.

8.2 Use automatic particle counters calibrated per Reference No. 5.

8.3 Use AC Fine Test Dust (ACFTD).

8.4 Use sample bottles containing less than 1.5 particles per millilitre per bottle volume greater than 10 micrometres as qualified per Reference No. 6.

8.5 Use test fluid conforming to Mil-H-5606, NATO symbol H-515, or DTD585B Hydraulic Fluid Specification.

NOTE. – The use of this carefully controlled hydraulic fluid assures greater reproducibility of results and is based upon current practices, other accepted filter standards, and its worldwide availability.

8.6 Use a filter performance test circuit comprised of a "filter test system" and a "contaminant injection system" as typified in Figure 1.

8.6.1 The filter test system consists of:

8.6.1.1 A reservoir constructed with a conical bottom displaying an included angle of not more than 90° with the entering oil diffused below the fluid surface.

NOTE. – This is to eliminate horizontal surfaces, to minimize contaminant settling, and to reduce air entrainment.

8.6.1.2 A hydraulic pump which is essentially insensitive to contaminant at the operating pressures.

WARNING. – Pumps exhibiting excessive flow pulses will cause erroneous results.

8.6.1.3 A system clean-up filter capable of maintaining an initial system contamination level of less than 15 particles per millilitre greater than 10 micrometres.

NOTE. – this cleanliness level assures that the test fluid is clean enough so as not to significantly influence the test results.

8.6.1.4 Pressure gages, temperature indicator, and controller and flow meter.

8.6.1.5 Pressure taps in accordance with Reference No. 7.

8.6.1.6 A turbulent sampling means located upstream and downstream of the test filter. Sample per Reference No. 8 or any other ISO-approved sampling method.

8.6.1.7 Interconnecting lines which insure that turbulent mixing conditions exist throughout the filter test system and that contaminant traps, silting areas and combinations of cyclonic separation zones and quiescent chambers are avoided.

8.6.2 The contaminant injection system consists of:

8.6.2.1 A reservoir constructed with a conical bottom displaying an included angle of not more than 90° with the entering oil diffused below the fluid surface.

NOTE. – This is to eliminate horizontal surfaces, to minimize contaminant settling and air entrainments.

8.6.2.2 A system clean-up filter capable of providing an initial contamination level of less than 1000 particles per millilitre greater than 10 micrometres and a gravimetric level less than 2 percent of the calculated level at which the test is being conducted.

8.6.2.3 A hydraulic pump (centrifugal or other types which do not alter the contaminant particle size distribution).

8.6.2.4 A sampling means for the extraction of a small flow (injection flow) from a point in the contaminant injection system where active circulation of fluid exists. Sample per Reference No. 8.

8.6.2.5 A variable rate slurry metering pump capable of injecting contaminant into the filter test system.

8.6.2.6 Interconnecting lines which insure that turbulent mixing conditions exist throughout the contaminant injection system and that contaminant traps, silting areas and combinations of cyclonic separation zones and quiescent chambers are not present. In particular, turbulent mixing conditions must exist throughout the length of the line conducting the injection fluid.

8.7 Use membranes and associated laboratory equipment suitable for conducting the double membrane gravimetric method per Reference No. 9.

9. TEST CONDITIONS ACCURACY

Set up and maintain equipment accuracy within the limits of Table 1.

Table E1. Test Conditions Accuracy.

Test Condition	SI Unit	US Unit	Maintain Within (±) of True Value
Flow	ℓ/min	USGPM	2%
Pressure	bar	psi	2%
Temperature	°C	°F	38°±2°C (100°±4°F)
Volume	ℓ	US Gal.	2%

10. FILTER PERFORMANCE TEST CIRCUIT VALIDATION PROCEDURES

NOTE. – These validation procedures reveal the effectiveness of the filter performance test circuit to maintain contaminant entrainment and/or prevent contaminant size modification.

10.1 Validation of Filter Test System

10.1.1 Validate initially at the minimum flow that the filter test system will be operated.

NOTE. – Install a filter housing or a conduit in place of a filter housing during validation.

10.1.2 Adjust the total test system fluid volume to be numerically equal to the minimum volume flow per minute value.

NOTE. – This is the volume to flow ratio required by the filter test procedure. Refer to Clause 11.3.7.

10.1.3 Contaminate the system fluid to a calculated gravimetric level of 10 milligrams per litre using AC Fine Test Dust.

NOTE. – This contamination level is below the saturation limitations of current automatic particle counters with high concentration sensors.

10.1.4 Circulate the fluid in the test system for one hour, extracting fluid samples at 15, 30, 45, and 60 minutes.

10.1.5 Analyze the four fluid samples and record three cumulative particle counts at 10 and 20 micrometres for each sample.

10.1.6 Accept the validation test only if:

10.1.6.1 The average of all three particle counts obtained for a given size from each sample does not deviate more than ten percent from the average particle counts of that size from all samples.

10.1.6.2 The average of all particle counts per millilitre at 10 micrometres is not less than 1200 nor more than 1800.

10.1.6.3 The particle counts per millilitre at 20 micrometres are not less than 200 nor more than 300.

10.2 Validation of Contaminant Injection System

10.2.1 Validate at the maximum gravimetric level and maximum injection circuit volume to be used; refer to Clauses 11.3.3 and 11.3.4.

10.2.2 Add the required quantity of contaminant in slurry form to the injection system fluid and circulate for two hours.

10.2.3 Extract fluid samples at the point where the injection fluid is discharged into the filter test system at 30, 60, 90, and 120 minutes and analyze each sample gravimetrically per Reference No. 9.

10.2.4 Accept the validation test only if the gravimetric level of each sample is within $\pm$ 10 percent of the average of the four samples and $\pm$ 10 percent of the known gravimetric value.

11. PRELIMINARY PREPARATION

11.1 Filter Selection Procedure

11.1.1 Determine the AQL to be used from Table 2 based upon the acceptable quality limit (AQL) as required by the associated specification.

11.1.2 Determine the filter sample size code letter from Table 3 for inspection level IV based upon the total lot size.

11.1.3 Use Table 4, the AQL, and the sample size code letter to determine the sample size.

11.1.4 Select the required number of sample filters at random for evaluation from the total lot.

Table E2.

AQL Conversion Table

For specified AQL values falling within these ranges	Use this AQL value
—— to 0.049	0.04
0.050 to 0.069	0.065
0.070 to 0.109	0.10
0.110 to 0.164	0.15
0.165 to 0.279	0.25
0.280 to 0.439	0.40
0.440 to 0.699	0.65
0.700 to 1.09	1.0
1.10 to 1.64	1.5
1.65 to 2.79	2.5
2.80 to 4.39	4.0
4.40 to 6.99	6.5
7.00 to 10.9	10.0
11.00 to 16.4	15.0

Table E3.

Sample Size Code Letters[1]

Lot Size	Inspection Levels				
	I	II	III	IV	V
3 to 8	B	B	B	B	C
9 to 15	B	B	B	B	D
16 to 25	B	B	B	C	E
26 to 40	B	B	B	D	F
41 to 65	B	B	C	E	G
66 to 110	B	B	D	F	H
111 to 180	B	C	E	G	I
181 to 300	B	D	F	H	J
301 to 500	C	E	G	I	K
501 to 800	D	F	H	J	L
801 to 1,300	E	G	I	K	L
1,301 to 3,200	F	H	J	L	M
3,201 to 8,000	G	I	L	M	N
8,001 to 22,000	H	J	M	N	O
22,001 to 110,000	I	K	N	O	P
110,001 to 550,000	I	K	O	P	Q
550,001 and over	I	K	P	Q	Q

[1]Sample size code letters given in body of table are applicable when the indicated inspection levels are to be used.

Table E4.

Standard Deviation Method

Master Table For Normal and Tightened Inspection for Plans Based on Variability Unknown
(Single Specification Limit—Form 1)

Sample size code letter	Sample size	Acceptable Quality Levels (normal inspection)													
		.04	.065	.10	.15	.25	.40	.65	1.00	1.50	2.50	4.00	6.50	10.00	15.00
		k	k	k	k	k	k	k	k	k	k	k	k	k	k
B	3	↓	↓	↓	↓	↓	↓	↓	▼	▼	1.12	.958	.765	.566	.341
C	4	↓	↓	↓	↓	↓	↓	↓	1.45	1.34	1.17	1.01	.814	.617	.393
D	5	↓	↓	↓	↓	↓	↓	1.65	1.53	1.40	1.24	1.07	.874	.675	.455
E	7	↓	↓	↓	↓	2.00	1.88	1.75	1.62	1.50	1.33	1.15	.955	.755	.536
F	10	↓	↓	↓	2.24	2.11	1.98	1.84	1.72	1.58	1.41	1.23	1.03	.828	.611
G	15	2.64	2.53	2.42	2.32	2.20	2.06	1.91	1.79	1.65	1.47	1.30	1.09	.886	.664
H	20	2.69	2.58	2.47	2.36	2.24	2.11	1.96	1.82	1.69	1.51	1.33	1.12	.917	.695
I	25	2.72	2.61	2.50	2.40	2.26	2.14	1.98	1.85	1.72	1.53	1.35	1.14	.936	.712
J	30	2.73	2.61	2.51	2.41	2.28	2.15	2.00	1.86	1.73	1.55	1.36	1.15	.946	.723
K	35	2.77	2.65	2.54	2.45	2.31	2.18	2.03	1.89	1.76	1.57	1.39	1.18	.969	.745
L	40	2.77	2.66	2.55	2.44	2.31	2.18	2.03	1.89	1.76	1.58	1.39	1.18	.971	.746
M	50	2.83	2.71	2.60	2.50	2.35	2.22	2.08	1.93	1.80	1.61	1.42	1.21	1.00	.774
N	75	2.90	2.77	2.66	2.55	2.41	2.27	2.12	1.98	1.84	1.65	1.46	1.24	1.03	.804
O	100	2.92	2.80	2.69	2.58	2.43	2.29	2.14	2.00	1.86	1.67	1.48	1.26	1.05	.819
P	150	2.96	2.84	2.73	2.61	2.47	2.33	2.18	2.03	1.89	1.70	1.51	1.29	1.07	.841
Q	200	2.97	2.85	2.73	2.62	2.47	2.33	2.18	2.04	1.89	1.70	1.51	1.29	1.07	.845
		.065	.10	.15	.25	.40	.65	1.00	1.50	2.50	4.00	6.50	10.00	15.00	
		Acceptable Quality Levels (tightened inspection)													

All AQL values are in percent defective.

↓ Use first sampling plan below arrow, that is, both sample size as well as k value. When sample size equals or exceeds lot size, every item in the lot must be inspected.

11.2 Insure that the test housing-filter arrangement will not allow fluid to bypass the filter elements to be evaluated.

11.3 Contaminant Injection System

11.3.1 Select an injection flow between .5 litres/minute and .25 litres/minute based upon number of filters to be tested, the test flow rate, and injection system volume.

11.3.2 Calculate minimum injection volume (ℓ) necessary to conduct the tests using the following equation:

$$V_i = 3 \text{ min.} \times (\text{Injection Flow, } \ell/\text{min})(\text{Total No. of Filters to Be Tested})$$

NOTES. –

1. The volume calculated above will assure a sufficient quantity of contaminated fluid to evaluate the test elements plus approximately 20 percent for adequate circulation throughout the test. Larger injection system volumes may be used.

2. A 0.5 ℓ/minute value of the injection flow will insure that the downstream sample flow expelled from the filter test system will not significantly influence the test results at the lower flow rate restriction given in the scope. Lower injection flow rates may be used provided that the base upstream gravimetric level of 10 mg/litre is maintained. An injection flow rate below 0.25 ℓ/minute is not recommended, due to silting characteristics and accuracy limitations.

11.3.3 Calculate the gravimetric level (γ', mg/litre) of the injection system fluid using the following equation:

$$\gamma' = \frac{(10 \text{ mg/litre})(\text{Test Flow, litres/minute})}{(\text{Injection Flow, litres/minute})}$$

11.3.4 Calculate the quantity of contaminant (ω, grams) needed for the contaminant injection system by the following equation:

$$\omega \text{ (grams)} = \frac{(\gamma', \text{ mg/}\ell)(\text{Injection System Volume, litres})}{1000}$$

11.3.5 Adjust the injection flow rate at stabilized temperature to within $\pm$ 5 percent of the value selected.

11.3.6 Adjust the total volume of the contaminant injection system to the value determined in Clause 11.3.2.

11.3.7 Circulate the fluid in the contaminant injection system through its system cleanup filter until a contamination level of less than 1000 particles per millilitre greater than 10 micrometres and a gravimetric level less than two percent of the value determined in Clause 11.3.3 are attained.

11.3.8 Bypass the system cleanup filter after the required initial contamination level has been achieved.

11.3.9 Add in slurry form the quantity (grams of contaminant determined in Clause 11.3.4) to the injection system reservoir.

11.3.10 Circulate the fluid in the injection system for a minimum of 15 minutes to thoroughly disperse the contaminant.

11.4 Filter Test System

11.4.1 Install the filter housing (without the test element) in the filter test system.

11.4.2 Circulate the fluid in the filter test system at rated flow and a stabilized test temperature of $38° \pm 2°C$ ($100° \pm 4°F$).

11.4.3 Adjust the total fluid volume of the filter test system (exclusive of the system cleanup filter circuit) such that it is numerically equal to the designated test volume flow through the filter per minute.

11.4.4 Circulate the fluid in the filter test system through the system cleanup filter until a contamination level of less than 15 particles per millilitre greater than 10 micrometres is attained.

NOTE. – The time required to achieve this contamination level is directly proportional to particle separation capability of the cleanup filter utilized.

11.4.5 Select and install the proper lengths of capillary tubing upstream and downstream of the test filter such that the initial upstream sample flow and the downstream sample flow are 0.5 ± 0.1 ℓ/minute. Maintain uninterrupted flow from the two sampling points during the entire test.

11.4.6 Return the sampling flows directly to the reservoir when sampling is not in progress.

12. FILTER PERFORMANCE TEST

12.1 Start system and stabilize temperature at 38°±2°C (100°±4°F).

12.2 Initial Contamination Level

12.2.1 Obtain a sample from upstream of the bypass loop.

12.2.2 Determine if contaminant level is less than 15 particles per millilitre greater than ten micrometres as set forth in 8.6.1.3.

12.2.3 Continue cleanup filtration if contaminant level is too high.

12.3 Adjust test system flow to equal rated filter flow.

12.4 Adjust load valve to insure adequate pressure to maintain proper flow through test system.

12.5 Shift test system flow to bypass loop.

12.6 Install filter to be tested.

12.7 Shift valve, forcing test flow (rated flow of filter) through filter test specimen.

12.8 Monitor flow for one minute.

12.9 Readjust flow if it varies more than ±2% from that set in 12.3.

12.10 Measure and record differential pressure across filter assembly.

12.11 Start injection metering pump and timer simultaneously. Fluid upstream should now be contaminated to ten milligrams per litre.

12.12 Upstream and downstream samples are taken after two minutes exposure to contaminated oil.

12.13 Stop metering pump.

12.14 Divert flow to bypass loop.

12.15 Remove filter test specimen.

12.16 Continue testing of similar sample filters by proceeding to 12.6 until the complete sample lot has been tested.

13. DETERMINATION OF LOT ACCEPTABILITY

13.1 Analyze samples taken in 12.12 by determining the number of particles greater than 10 and 30 micrometres by any ISO-approved counting method. (The use of on-line particle counting techniques is recommended for this purpose.)

13.2 Proceed to 13.6 if all upstream particle counts are within $\pm$5% of the recommended level.

13.3 Total upstream particle counts and divide by number of samples taken to determine average.

13.4 Determine corrected downstream count using the following equation:

$$(N_d/N_u) \quad N_{u_{avg}} \quad = \quad N_{d_{corrected}}$$

13.5 Note on all reports concerning this particular test that downstream counts were normalized.

13.6 Sum all downstream particle counts.

13.7 Determine correction factor by squaring the sum of 13.6 and dividing by sample size.

13.8 The corrected sum of squares is the difference between the sum of the measurements squared and the correction factor.

13.9 The variance is the corrected sum of squares (13.8) divided by one less than the sample size.

13.10 Estimate the lot standard deviation, the square root of the variance (13.9).

13.11 The sample mean is the sum of the measurements (13.6) divided by the sample size.

13.12 The specification upper limit is given in the filter specification.

13.13 A quantity for comparison is determined by taking the difference between the specification upper limit (13.12) and the sample mean (13.11) divided by the estimated standard deviation (13.10).

13.14 The acceptability constant is determined from Table 4.

13.15 The lot is accepted only if the quanity for comparison (13.13) is greater than the acceptability constant (13.14).

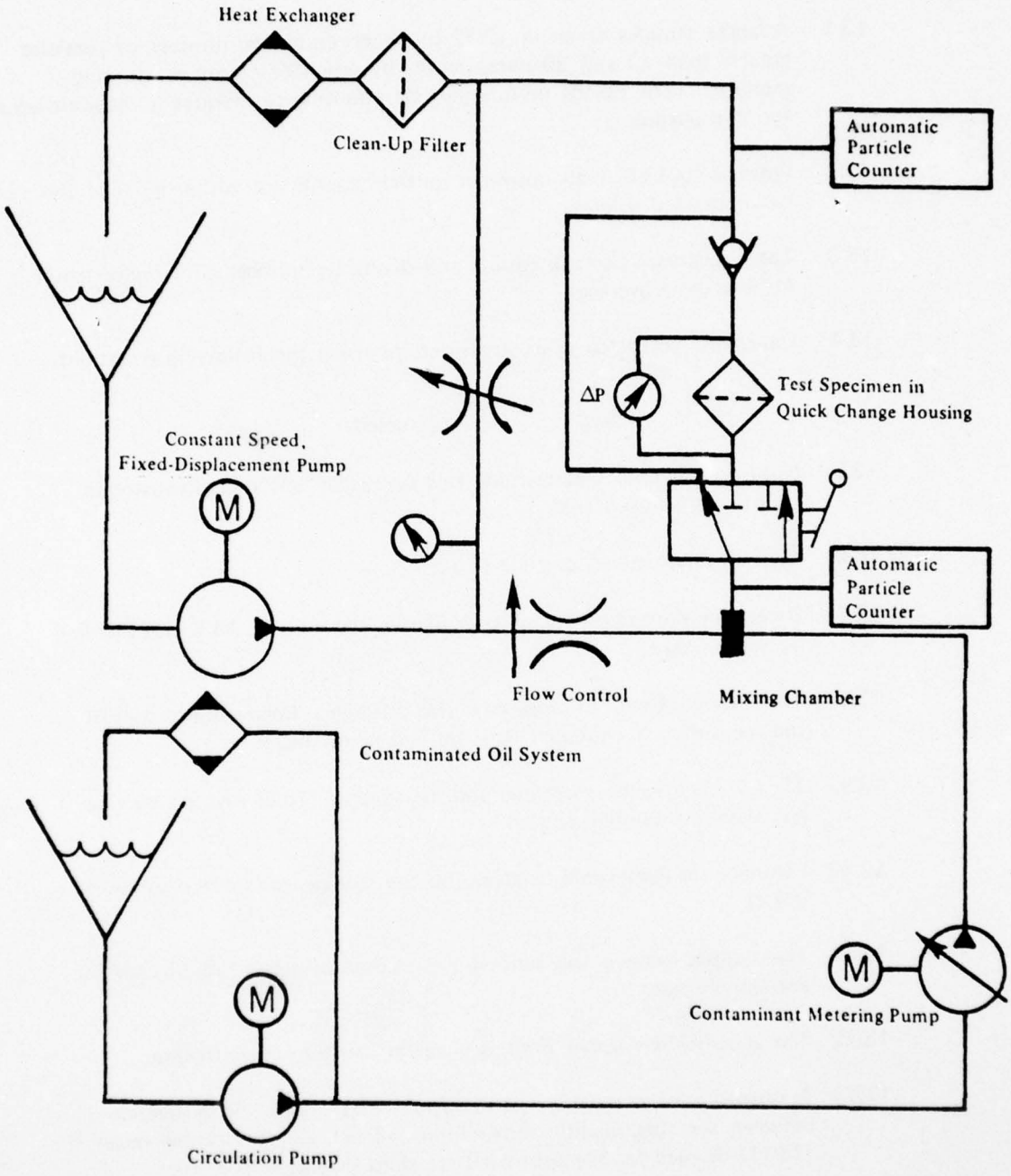


Fig. E-1. First Draft Proposal Test Schematic.

13. DETERMINATION OF LOT ACCEPTABILITY

13.1 Analyze samples taken in 12.12 by determining the number of particles greater than 10 and 30 micrometres by any ISO-approved counting method. (The use of on-line particle counting techniques is recommended for this purpose.)

13.2 Proceed to 13.6 if all upstream particle counts are within $\pm 5\%$ of the recommended level.

13.3 Total upstream particle counts and divide by number of samples taken to determine average.

13.4 Determine corrected downstream count using the following equation:

$$(N_d/N_u) \quad N_{u_{avg}} \quad = \quad N_{d_{corrected}}$$

13.5 Note on all reports concerning this particular test that downstream counts were normalized.

13.6 Sum all downstream particle counts.

13.7 Determine correction factor by squaring the sum of 13.6 and dividing by sample size.

13.8 The corrected sum of squares is the difference between the sum of the measurements squared and the correction factor.

13.9 The variance is the corrected sum of squares (13.8) divided by one less than the sample size.

13.10 Estimate the lot standard deviation, the square root of the variance (13.9).

13.11 The sample mean is the sum of the measurements (13.6) divided by the sample size.

13.12 The specification upper limit is given in the filter specification.

13.13 A quantity for comparison is determined by taking the difference between the specification upper limit (13.12) and the sample mean (13.11) divided by the estimated standard deviation (13.10).

13.14 The acceptability constant is determined from Table 4.

13.15 The lot is accepted only if the quanity for comparison (13.13) is greater than the acceptability constant (13.14).

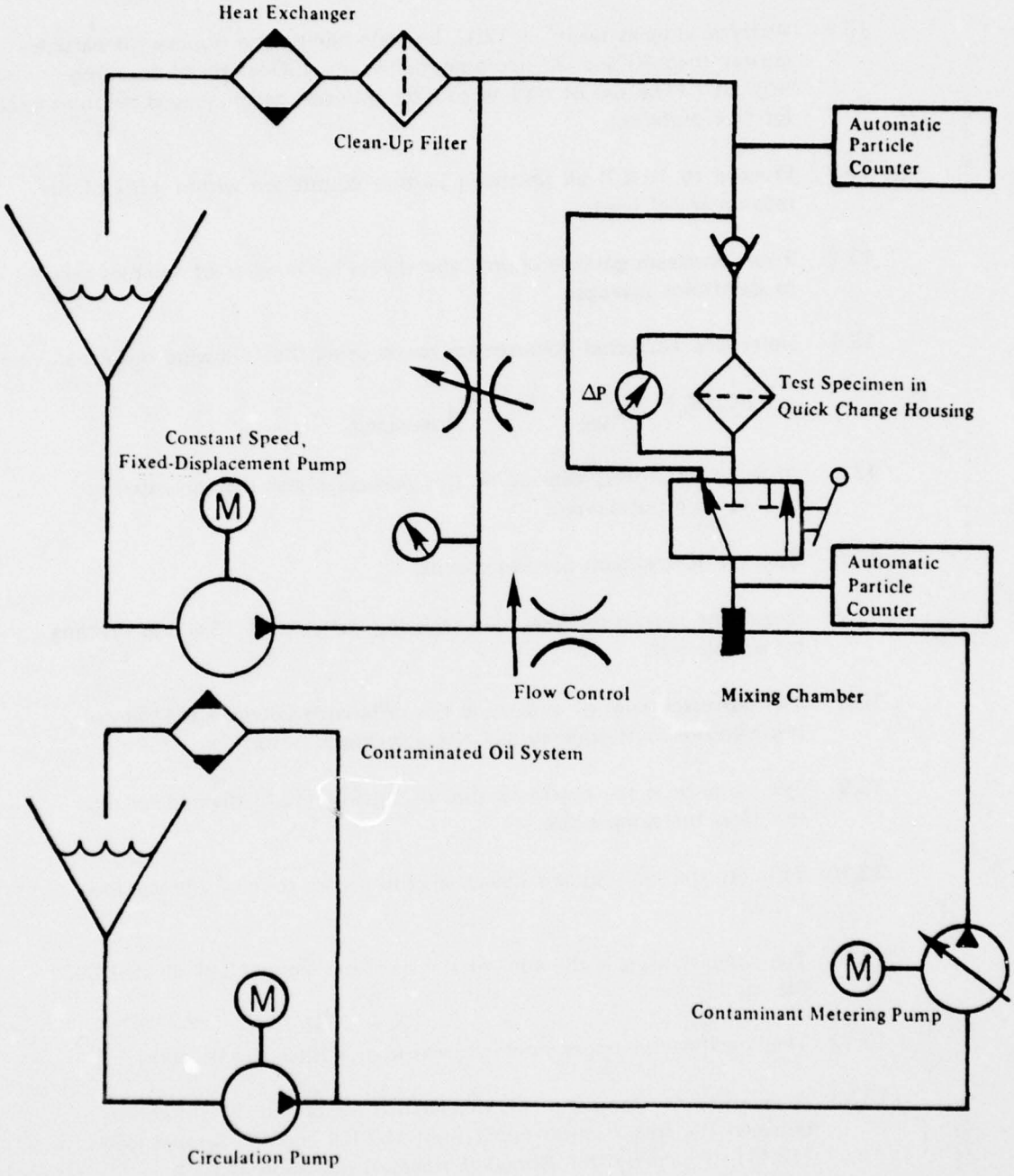


Fig. E-1. First Draft Proposal Test Schematic.

APPENDIX F

ACCELERATED HYDRAULIC FILTER ELEMENT MULTI-PASS TEST FOR QUALITY CONTROL

ACCELERATED HYDRAULIC FILTER ELEMENT MULTI-PASS TEST FOR QUALITY CONTROL

1. SCOPE

1.1 To include an accelerated multi-pass filtration performance test with continuous contaminant injection for fine hydraulic fluid power filter elements for quality control.

1.2 To include a procedure for determining the contaminant capacity, particulate removal characteristics and pressure loss.

1.3 To include a test currently applicable to hydraulic fluid power filter elements which exhibit a 10 micrometre filtration ratio less than 75.0, a final reservoir gravimetric level of less than 200 mg/ℓ, and that have a rated flow in the range between 4.0 ℓ/min and 600 ℓ/min (1 USGPM and 160 USGPM).

2. PURPOSE

To provide an accelerated test procedure which yields reproducible test data for appraising the filtration performance of a fine hydraulic fluid power filter element, for quality control.

3. TERMS AND DEFINITIONS

(For definitions of terms not defined, see Reference No. 1.)

Multi-Pass Test. A test which requires the recirculation of unaltered effluent fluid through the filter element.

Accelerated Multi-Pass Test. A multi-pass test with various, specified parameters modified in such a manner as to minimize test duration.

4. UNITS

4.1 The International System of Units (SI) is used in accordance with Reference No. 2.

4.2 Approximate conversions to "Customary US" units are given for information purposes, except where US practice conforms to SI usage.

5. LETTER SYMBOLS

(Letter symbols are in accordance with Reference No. 2.)

6. GRAPHIC SYMBOLS

Graphic symbols used are in accordance with Reference No. 3 and Reference No. 4. Where Reference No. 3 and Reference No. 4 are not in agreement, Reference No. 3 governs.

7. GENERAL PROCEDURE

7.1 Set up and maintain apparatus per sections 8 and 9.

7.2 Run all tests per sections 10, 11 and 12.

7.3 Analyze data from sections 10, 11 and 12 per sections 13, 14 and 16.

7.4 Present data from sections 12 and 14 per sections 15 and 20.

8. TEST EQUIPMENT

8.1 Use a suitable timer for measuring minutes and seconds.

8.2 Use automatic particle counter calibrated per Reference No. 5 or any ISO-approved counting method.

8.3 Use AC Fine Test Dust (ACFTD) dried at 110-150°C (230-300°F) for not less than one hour for quantities less than 200 grams.

8.4 Use sample bottles containing less than 1.5 particles per millilitre per bottle volume greater than 10 micrometres as qualified per Reference No. 6, or standardized inline particle counting techniques.

8.5 Use test fluid conforming to MIL-H-5606, NATO symbol H-515, or DTD585B Hydraulic Fluid Specification.

8.6 Use a filter performance test circuit comprised of a "filter test system" and a "contaminant injection system" as typified in Figure 1.

8.6.1 The filter test system consists of:

8.6.1.1 A reservoir constructed with a conical bottom displaying an included angle of not more than 90° with the entering oil diffused below the fluid surface.

3.6.1.2 A hydraulic pump which is essentially insensitive to contaminant at the operating pressures.

WARNING. - Pumps exhibiting excessive flow pulses will cause erroneous results.

8.6.1.3 A system clean-up filter capable of providing an initial system contamination level of less than 15 particles per millilitre greater than 10 micrometres.

8.6.1.4 Pressure gages, temperature indicator and controller and flow meter.

8.6.1.5 Pressure taps in accordance with Reference No. 7.

8.6.1.6 A turbulent sampling means located upstream and downstream of the test filter. Sample per Reference No. 8 or any other ISO-approved sampling method.

8.6.1.7 Interconnecting lines which insure that turbulent missing conditions exist throughout the filter test

system and that contaminant traps, silting areas and combinations of cyclonic separation zones and quiescent chambers are avoided.

8.6.2 The contaminant injection system consists of:

8.6.2.1 A reservoir constructed with a conical bottom displaying an included angle of not more than 90° with the entering oil diffused below the fluid surface.

8.6.2.2 A system clean-up filter capable of providing an initial contamination level of less than 1000 particles per millilitre greater than 10 micrometres and a gravimetric level less than 2 percent of the calculated level at which the test is being conducted.

8.6.2.3 A hydraulic pump (centrifugal or other types which do not alter the contaminant particle size distribution.)

8.6.2.4 A sampling means for the extraction of a small flow (injection flow) from a point in the contaminant injection system where active circulation of fluid exists. Sample per Reference No. 8.

8.6.2.5 Interconnecting lines which insure that turbulent mixing conditions exist throughout the contaminant injection system and that contaminant traps, silting

areas and combinations of cyclonic separation zones and quiescent chambers are not present. In particular, turbulent mixing conditions must exist throughout the length of the line conducting the injection fluid.

8.7 Use membranes and associated laboratory equipment suitable for conducting the double membrane gravimetric method per Reference No. 9.

9. TEST CONDITIONS ACCURACY

Set up and maintain equipment accuracy within the limits in Table F-1.

TABLE F-1. Test Conditions Accuracy.

Test Condition	SI Unit	US Unit	Maintain Within (±) of True Value
Flow	ℓ/min	USGPM	2%
Pressure	bar	psi	2%
Temperature	^{o}C	^{o}F	$38^{o} \pm 2^{o}C$ ($100^{o} \pm 4^{o}F$)
Volume	ℓ	US Gal.	2%

AD-A061 383 OKLAHOMA STATE UNIV STILLWATER FLUID POWER RESEARCH --ETC F/G 13/7
MERADCOM/OSU HYDRAULIC SYSTEM RELIABILITY PROGRAM.(U)
MAY 78 DAAK70-77-C-0097

UNCLASSIFIED OSU-FPRC-8M1 NL

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10. FILTER PERFORMANCE TEST CIRCUIT VALIDATION PROCEDURES

NOTE. - These validation procedures reveal the effectiveness of the filter performance test circuit to maintain contaminant entrainment and/or prevent contaminant size modification.

10.1 Validation of Filter Test System

10.1.1 Validate at the minimum flow that the filter test system will be operated.

NOTE. - Install a filter housing or a conduit in place of a filter housing during validation.

10.1.2 Adjust the total test system fluid volume to be numerically equal to one-fourth the minimum volume flow per minute value.

10.1.3 Contaminate the system fluid to a calculated gravimetric level of 5 milligrams per litre using AC Fine Test Dust.

10.1.4 Circulate the fluid in the test system for one hour, extracting fluid samples at 15, 30, 45 and 60 minutes.

10.1.5 Analyze the four fluid samples and record three cumulative particle counts at 10 and 20 micrometres for each sample.

10.1.6 Accept the validation test only if:

10.1.6.1 The average of all 3 particle counts obtained for a given size from each sample does not deviate more than 10 percent from the average particle counts of that size from all samples.

10.1.6.2 The average of all particle counts per millilitre at 10 micrometres is not less than 600 nor more than 900.

10.1.6.3 The particle counts per millilitre at 20 micrometres are not less than 100 nor more than 150.

10.2 Validation of Contaminant Injection System

10.2.1 Validate at the maximum gravimetric level and maximum injection circuit volume to be used; refer to clauses 11.2.2 and 11.2.3.

10.2.2 Add the required quantity of contaminant in slurry form to the injection system fluid and circulate for two hours.

10.2.3 Extract fluid samples at the point where the injection fluid is discharged into the filter test system at 30, 60, 90 and 120 minutes and analyze each sample gravimetrically per Reference No. 9.

10.2.4 Accept the validation test only if the gravimetric level of each sample is within ±10 percent of the average of the four samples and ±10 percent of the known gravimetric value.

11. PRELIMINARY PREPARATION

11.1 Test Filter Assembly

11.1.1 Insure that test fluid cannot bypass the filter element to be evaluated.

11.2 Contaminant Injection System

11.2.1 Using the specified level (3 to 50 mg/ℓ) as a base upstream gravimetric level, calculate the predicted test time (τ') in minutes by the following equation:

$$\tau' = \frac{\text{(Apparent Capacity of Filter Element, mg)}}{\text{(Specified level mg/ℓ) (Test Flow Rate, ℓ/min)}}$$

NOTE. - A base upstream gravimetric level in this range allows a minimum test duration to be achieved. This level for each type of filter tested shall be given in the filter specifications.

11.2.2 Calculate the minimum required operating injection system volume (σ, litres) which is compatible with the above predicted test time (τ') and a value for the injection flow of 0.5 ℓ/min using the following equation:

$$\sigma = 1.2 \text{ } (\tau', \text{ minutes}) \text{ (Injection Flow, ℓ/min)}$$

NOTES -

1. The volume calculated above will assure a sufficient quantity of contaminated fluid to load the test element plus 20 percent for adequate circulation throughout the test. Larger injection system volumes may be used.
2. The 0.5 ℓ/min value of the injection flow insure that the downstream sample flow expelled from the filter test system will not significantly influence the test results at the lower flow rate restriction given in the scope. Lower injection flow rates may be used provided that the base upstream gravimetric level specified is maintained. An injection flow rate below 0.25 ℓ/min is not recommended due to silting characteristics and accuracy limitation.

11.2.3 Calculate the gravimetric level (γ', mg/ℓ) of the injection system fluid using the following equation:

$$\gamma' = \frac{\text{(Specified level mg/ℓ) (Test flow, ℓ/min)}}{\text{(Injection Flow, ℓ/min)}}$$

11.2.4 Calculate the quantity of contaminant (ω, grams) needed for the contaminant injection system by the following equation:

$$\omega \text{ (grams)} = \frac{(\gamma', \text{ mg/}\ell) \text{ (Injecion System Volume, litres)}}{1000}$$

11.2.5 Adjust the injection flow rate at stabilized temperature to within ±5 percent of the value selected in clause 11.2.2 and maintain throughout the test.

11.2.6 Adjust the total volume of the contaminant injection system to the value determined in clause 11.2.2.

11.2.7 Circulate the fluid in the contaminant injection system through its system clean-up filter until a contamination level of less than 1000 particles per millilitre greater than 10 micrometres and a gravimetric level less than 2 percent of the value determined in clause 11.2.3 are attained.

11.2.8 Bypass the system clean-up filter after the required initial contamination level has been achieved.

11.2.9 Add in slurry form the quantity (grams of contaminant determined in clause 11.2.4) to the injection system reservoir.

11.2.10 Circulate the fluid in the injection system for a minimum of 15 minutes to thoroughly disperse the contaminant.

11.3 Filter Test System

11.3.1 Install the filter housing (without the test element) in the filter test system.

11.3.2 Circulate the fluid in the filter test system at rated flow and a stabilized test temperature of $38^{\circ} \pm 2^{\circ}C$ ($100^{\circ} \pm 4^{\circ}F$) and record the pressure drop of the empty filter housing.

11.3.3 Adjust the total fluid volume of the filter test system (exclusive of the system clean-up filter circuit) such that it is numerically equal to one-fourth the designated test volume flow through the filter per minute.

11.3.4 Circulate the fluid in the filter test system through the system clean-up filter until a contamination level of less than 15 particles per millilitre greater than 10 micrometres is attained, if the filter has a specified Beta 10 greater than 2.0. For filters with specified Beta 10 less than or equal to 2.0, 150 particles per millilitre greater than 10 micrometres is acceptable.

NOTE. - The time required to achieve this contamination level is directly proportional to particle separation capability of the clean-up filter utilized.

11.3.5 Select and install the proper lengths of capillary tubing upstream and downstream of the test filter such that the initial upstream sample flow is 0.3 ± 0.05 ℓ/min and the downstream sample flow is within 5 percent of the injection flow. Maintain uninterrupted flow from the two sampling points during the entire test. If inline particle counting techniques are used, install per applicable standard, or specification.

11.3.6 Return the sampling flow upstream of the test filter directly to the reservior when bottle sampling is not in progress.

11.3.7 Collect the sampling flow downstream of the test filter outside the filter test system in order to assist in maintaining a constant system volume which should be kept within 15 percent of the required system volume.

12. FILTER PERFORMANCE TEST

12.1 Install the filter element into its housing and subject the assembly to the specified test condition test flow and test temperature of $38^{\circ} \pm 2^{\circ}C$ ($100^{\circ} \pm 4^{\circ}F$) and reaffirm fluid level.

12.2 Measure and record the clean assembly pressure drop. Calculate and record the clean element pressure drop. Clean assembly minus the housing pressure drop measured in clause 11.3.2.

12.3 Calculate the pressure drops corresponding to increases of 5, 10, 20, 40, 80 and 100 percent of the net pressure drop. (Terminal pressure drop minus clean element pressure drop.)

12.4 Obtain a sample or inline particle count from upstream of the test filter element to determine the initial system contamination level.

NOTE. - Take all samples in such a manner so as to minimize the aeration of the fluid sample.

12.5 Obtain a sample from the contaminant injection system.

12.6 Measure and record the injection flow rate.

12.7 Initiate the filter test as follows:

12.7.1 Bypass the system clean-up filter

12.7.2 Allow the injection flow to enter the filter test system reservoir

12.7.3 Start the timer

12.7.4 Start the downstream sample flow

12.8 Record the test time (minutes) for the pressure drop across the filter assembly to increase by 5 percent of the net pressure drop.

12.9 Extract upstream and downstream samples or perform inline particle counts simultaneously when the pressure drop across the filter assembly has increased by 10 ± 1 percent of the net pressure drop. Record the test time to reach the 10 percent point.

NOTE. - Use identical sample time of not more than 30 seconds for both upstream and downstream samples. Since the sampling procedure requires the sample volume to be within 50 to 90 percent of the sample bottle volume, more than one size sample bottle may be required.

12.10 Repeat clause 12.9 at an increase of 40 ± 1 percent of the net pressure drop.

12.11 Record the test time (τ, minutes) for the pressure drop across the filter assembly to increase by 100 percent of the net pressure drop.

12.12 Continue injection to collapse or maximum pressure specified. This step is optional, refer to filter specification.

12.13 Conclude the test by stopping the flow to the test filter.

12.14 Obtain a fluid sample from the contaminant injection system if this is the final filter of a series, if the system will be shut down before testing of like filters continues, or, if more than 1 hour will elapse before the

next filter is tested.

12.15 Measure and record the injection flow rate.

13. DATA ACCURACY

Select and maintain instrumentation so that data accuracy is within the limits in Table 2, unless otherwise specified.

TABLE F-2. DATA ACCURACY

Quantity	SI Unit	US Unit	Accuracy Within (±) of True Value
Injection Flow Rate	ℓ/min	USGPM	5%
Base Upstream Gravimetric Level	mg/ℓ	- - -	1 mg/ℓ

14. CALCULATIONS

14.1 Analyze the samples extracted from the filter test system by determining the number of particles per millilitre greater than 10, 20, 30 and 40 mircometres with an automatic particle counter calibrated per Reference No. 5 or any ISO-approved counting method. (Other particle sizes may be analyzed, refer to specification.)

NOTE. - Care should be taken to dilute samples appropriately to avoid exceeding the saturation limit

of the counting method determined by the approved calibration procedure.

14.1.1 Obtain a minimum of three particle counts for each fluid sample and calculate the arithmetic average for each size range counted.

14.1.2 Accept the test only if the number of particles per millilitre greater than 10 micrometres in the initial sample from the filter test system is less than 15, or 150 if the specified beta is 2.0 or less.

14.2 Conduct a gravimetric analysis per Reference No. 9 on the samples extracted from the contaminant injection system.

14.2.1 Calculate the average (γ) of the two gravimetric levels from the injection system. (The sample taken immediately before this test and that taken after, per 12.14 or 12.5.)

14.2.2 Accept the test only if the gravimetric level of each sample is within ± 10 percent of this average.

14.3 Calculate and record the injection flow rate by averaging the measurements taken at the beginning and end of the test.

14.3.1 Accept the test only if this value is equal to the selected value ± 5 percent.

14.4 Calculate and record the actual base upstream gravimetric level by multiplying the average injection gravimetric level (γ, mg/ℓ) by the average injection flow rate (ℓ/min) per clause 14.3 and dividing by the test flow (ℓ/min).

14.4.1 Accept the test only if this value is equal to specified base upstream gravimetric level ±10%.

14.5 Calculate the filtration ratio as defined in Section 3.

14.5.1 Record these calculated ratios.

14.5.2 Record the minimum filtration ratio.

15.1 Report the following minimum information for filter elements evaluated using this recommended standard:

15.2 Using the actual test time (τ) to reach the terminal pressure drop, the average gravimetric level (γ) of the injection stream, and the injection flow rate, calculate the filter element ACFTD capacity (α) using the following equation:

$$\alpha, \text{grams} = \frac{(\gamma, \text{mg}/\ell)(\text{Injection Flow Rate}, \ell/\text{min})(\tau, \text{min})}{1000}$$

NOTE. - USGPM can be converted to ℓ/min by multiplying by 3.7854.

15.2.1 Record the ACFTD capacity.

15.3 Report the values of the gravimetric levels obtained in clause 14.2.

15.4 Plot the average particle count points obtained from the upstream and downstream samples extracted at the 80 percent sample point on the Filter Performance Chart shown in Figure 2.

15.5 Have available a record of all the following minimum test data in all test reports referencing this recommended standard:

15.5.1 All physical values pertaining to the test.

15.5.2 All additional provisions or modifications pertaining to the test.

16. CRITERIA FOR ACCEPTANCE

16.1 Compare the minimum filtration ratio (β_{10}) with the designated value.

16.2 Compare the filter element ACFTD capacity (α) with the designated value.

16.3 No visual evidence of filter element damage as a result of performing this test.

17. SUMMARY OF DESIGNATED INFORMATION

The following designated information is needed when applying this recommended standard to a particular application or use:

17.1 Filter element test flow

17.2 Terminal pressure drop

17.3 The acceptable minimum value of filtration (β_{10}).

17.4 The acceptable minimum value of the filter element ACFTD capacity (α).

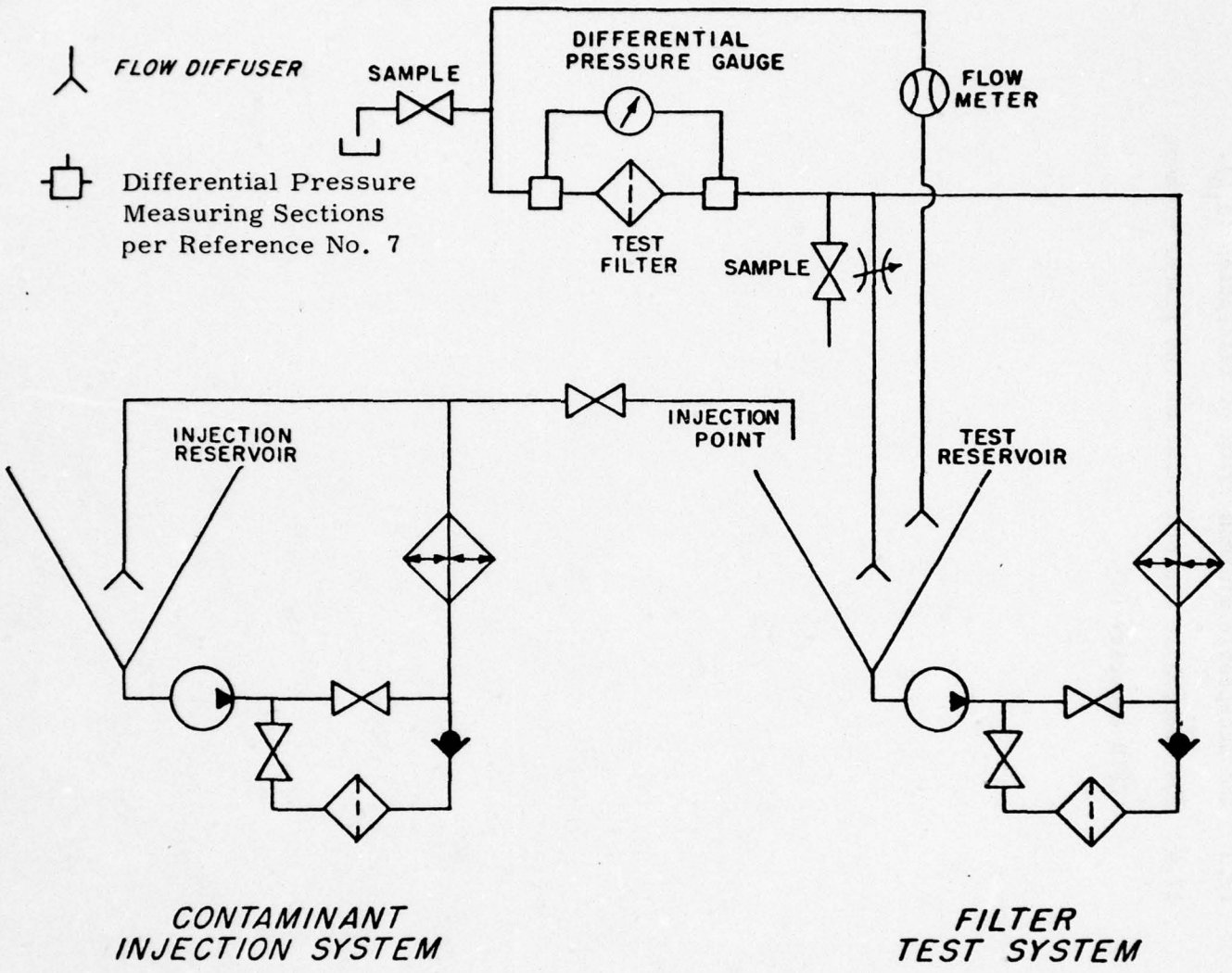


FIG. F-1. Typical filter performance test schematic.

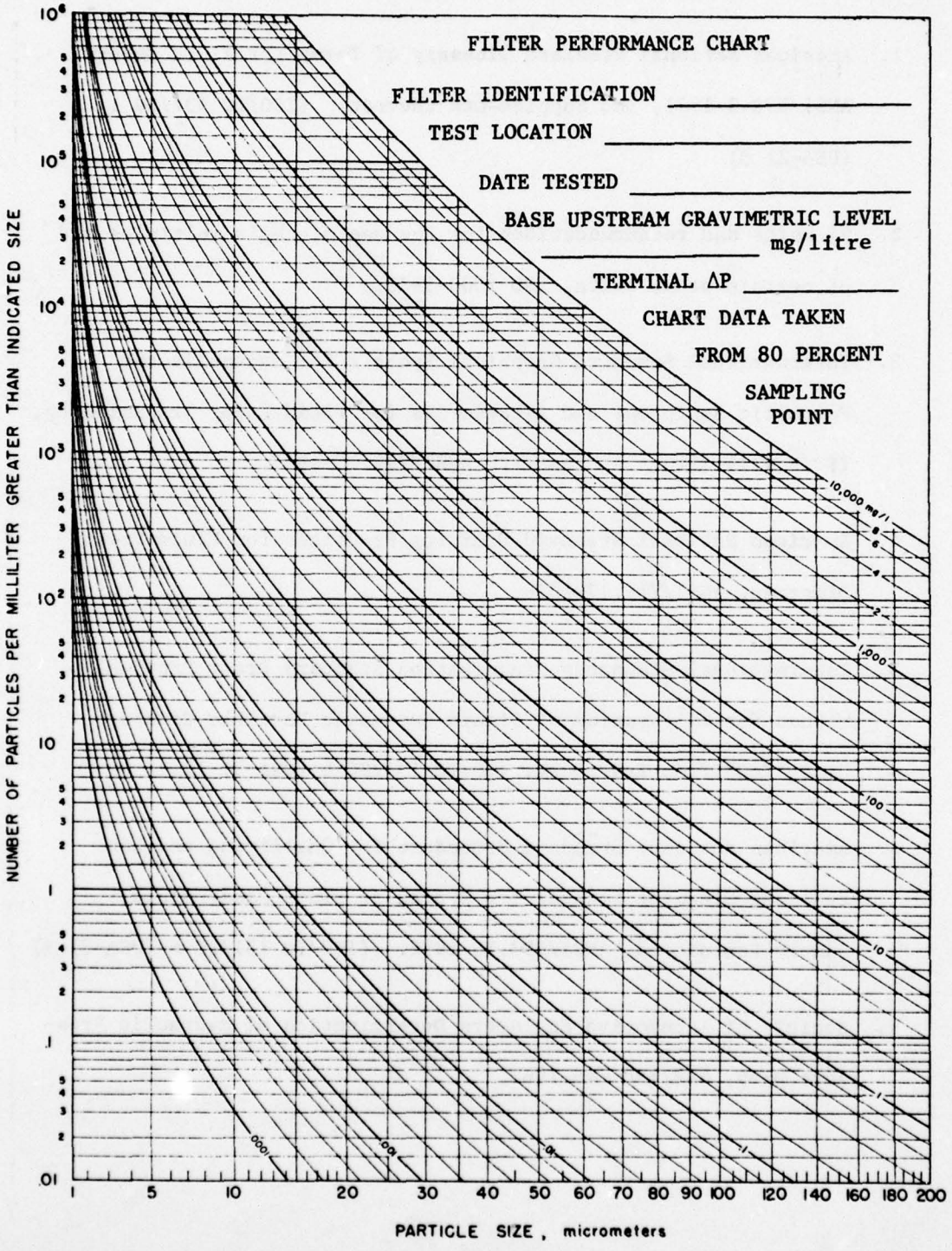


Fig. F-2. Filter Performance Chart

REFERENCES

1. American National Standard Glossary of Terms for Fluid Power, ANSI/B93.2-1971, and Supplements thereto. (ISO/TC 131/SC 1 (USA-2) 3)

2. SI units and recommendations for the use of their multiples and of certain other units, ISO 1000-1973.

3. International Standard Graphical Symbols for Hydraulic and Pneumatic Equipment and Accessories for Fluid Power Transmission, ISO/R 1219-1970. Agrees with ANSI/Y32.10-1967.

4. American National Standard Drafting Practices for Fluid Power Diagrams, ANSI/Y14.17-1966.

5. International Standards Organization Standard Draft Proposal Method for Calibration of Liquid Automatic Particle Counters Using "AC" Fine Test Dust, ISO/TC 131/SC 6 (USA-9) 12.

6. American National Standard Procedure for Qualifying and Controlling Cleaning Methods for Hydraulic Fluid Power Fluid Sample Containers, ANSI/B93.20-1972. (ISO/TC 131/SC 6 (USA-8) 9)

7. Society of Automotive Engineers Determination of Hydraulic Pressure Drop, SAE/ARP 24B-1968.

8. American National Standard Method for Extracting Fluid Samples from the Lines of an Operating Hydraulic Fluid Power System (for Particulate Contamination Analysis), ANSI/B93.19-1972. (ISO/TC 131/SC 6 (USA-2) 3)

9. Society of Automotive Engineers Procedure for the Determination of Particulate Contamination in Hydraulic Fluids by the Control Gravimetric Procedure, SAE/ARP 785-1963.

APPENDIX G

FILTER TEST DATA

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FILTER ELEMENT MULTI-PASS-TEST REPORT SHEET

FILTER ID:MERADCOM/FPRC QC TYPE 1 DATE: 05APR78
LAB: FPRC-OSU FPRC NO. 438 FLOW RATE: 75.0 LPM R.H.: 40.0 %
BUBBLE POINT:34.3CM WATER ELEMENT SIZE (LXD) 22.9 X 7.0 CM.

DIFFERENTIAL PRESSURES,KPA

TERMINAL:	275.9
HOUSING:	31.0
CLEAN ASSEMBLY:	50.3
CLEAN ELEMENT:	19.3
NET:	256.6

SYSTEM VOLUME(L): 18.8

INJECTION	INITIAL	FINAL	AVERAGE
FLOW (LPM)	0.161	0.160	0.160
GRAV.(MG/L)	4678.01	4710.01	4694.01

UPSTREAM GRAVIMETRIC LEVELS(MG/L): BASE 10.04, FINAL 32.50

% NET PRESS	2.5%	5%	10%	20%	40%	80%	100%
ASSEMBLY PRESS	56.8	63.2	76.0	101.7	153.0	255.6	306.9
GRAMS ADDED	3.8	5.5	7.2	8.5	9.6	11.2	11.6
TEST TIME,MIN.	5.1	7.4	9.6	11.3	12.9	14.8	15.4

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE POINT		> 5 MIC	> 10 MIC	> 20 MIC	> 30 MIC	> 40 MIC
INITIAL		24.10	10.10	3.10	1.20	0.50
2MIN	UP	5127.00	1444.00	267.60	87.20	38.90
	DOWN	75.30	19.90	3.93	1.47	0.70
	BETA	68.09	72.56	68.09	59.32	55.57
10 %	UP	5390.00	1494.00	547.90	89.30	37.60
	DOWN	235.60	56.50	12.00	4.00	1.60
	BETA	22.88	22.47	45.66	22.32	23.50
20 %	UP	5592.00	2728.00	288.20	94.70	39.30
	DOWN	212.30	52.70	10.73	3.33	1.53
	BETA	26.34	51.76	26.86	28.44	25.69
40 %	UP	5431.00	1493.00	270.10	90.50	37.60
	DOWN	287.60	71.00	15.23	5.30	2.17
	BETA	18.88	21.03	17.73	17.08	17.33
80 %	UP	5399.00	1502.00	271.00	86.30	34.60
	DOWN	421.50	110.50	22.50	0.00	0.00
	BETA	12.81	13.59	12.04	*********	*********
AVERAGE BETA		22.48	24.13	29.16	27.86	28.38
MINIMUM BETA		12.81	13.59	12.04	17.08	17.33

ACFTD CAPACITY(GRAMS): APPARENT 11.6, RETAINED 11.0

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FILTER ELEMENT MULTI-PASS-TEST REPORT SHEET

FILTER ID:MERADCOM/FPRC QC TYPE 1 DATE: 04MAY78
LAB: FPRC-OSU FPRC NO. 439 FLOW RATE: 75.0 LPM R.H.: 40.0 %
BUBBLE POINT: 30.5CM WATER ELEMENT SIZE (LXD) 22.9 X 7.0 CM.

DIFFERENTIAL PRESSURES,KPA

TERMINAL:	275.9
HOUSING:	31.0
CLEAN ASSEMBLY:	30.3
CLEAN ELEMENT:	19.3
NET:	256.6
SYSTEM VOLUME(L):	18.8

INJECTION	INITIAL	FINAL	AVERAGE
FLOW (LPM)	0.161	0.162	0.161
GRAV.(MG/L)	4868.0	4833.0	4850.5

UPSTREAM GRAVIMETRIC LEVELS(MG/L): BASE 10.44, FINAL 7.35

% NET PRESS	2.5%	5%	10%	20%	40%	80%	100%
ASSEMBLY PRESS	56.8	63.2	76.0	101.7	153.0	255.6	306.9
GRAMS ADDED	4.4	6.1	7.6	9.0	10.6	11.8	12.4
TEST TIME,MIN.	5.6	7.8	9.8	11.5	13.5	15.1	15.8

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE POINT		> 5 MIC	> 10 MIC	> 20 MIC	> 30 MIC	> 40 MIC
INITIAL		*********	*********	*********	*********	*********
2MIN	UP	4997.00	1488.00	272.00	58.20	39.07
	DOWN	76.13	23.90	4.60	1.40	0.73
	BETA	65.64	62.26	59.13	63.00	53.52
10 %	UP	5267.00	1563.00	282.90	87.47	36.93
	DOWN	140.20	41.17	7.79	3.00	1.60
	BETA	37.57	37.96	36.32	29.16	24.33
20 %	UP	5229.00	1544.00	268.40	88.13	40.40
	DOWN	193.80	54.46	9.93	3.66	1.53
	BETA	26.98	28.35	27.03	24.76	26.41
40 %	UP	5437.00	1603.00	277.90	91.60	39.93
	DOWN	297.90	83.63	15.40	4.97	2.07
	BETA	18.25	19.17	18.05	18.43	19.29
80 %	UP	5405.00	1629.00	301.40	102.10	45.93
	DOWN	715.20	201.00	38.33	0.00	0.00
	BETA	7.56	8.10	7.90	*********	*********
AVERAGE BETA		24.24	25.18	24.15	31.57	28.72
MINIMUM BETA		7.56	8.10	7.90	18.43	19.29

ACFTD CAPACITY(GRAMS): APPARENT 12.4, RETAINED 12.2

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FILTER ELEMENT MULTI-PASS-TEST REPORT SHEET

FILTER ID:MERADCOM/FPRC QC TYPE 1 DATE: 05APR78
LAB: FPRC-OSU FPRC NO. 440 FLOW RATE: 75.0 LPM R.H.: 40.0 %
BUBBLE POINT:33.0CM WATER ELEMENT SIZE (LXD) 22.9 X 6.3 CM.

DIFFERENTIAL PRESSURES,KPA

TERMINAL:	275.9
HOUSING:	31.0
CLEAN ASSEMBLY:	50.3
CLEAN ELEMENT:	19.3
NET:	256.6
SYSTEM VOLUME(L):	18.8

INJECTION	INITIAL	FINAL	AVERAGE
FLOW (LPM)	0.161	0.161	0.161
GRAV.(MG/L)	4810.0	4805.0	4807.5

UPSTREAM GRAVIMETRIC LEVELS(MG/L): BASE 10.32, FINAL 17.00

% NET PRESS	2.5%	5%	10%	20%	40%	80%	100%
ASSEMBLY PRESS	56.8	63.2	76.0	101.7	153.0	255.2	306.9
GRAMS ADDED	4.3	5.9	7.6	9.0	10.3	11.7	12.2
TEST TIME,MIN.	5.5	7.6	9.8	11.6	13.4	15.2	15.8

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE POINT		> 5 MIC	> 10 MIC	> 20 MIC	> 30 MIC	> 40 MIC
INITIAL		23.40	10.20	5.90	3.60	2.50
2MIN	UP	5002.00	1366.00	252.00	85.20	38.10
	DOWN	73.77	19.10	3.90	1.40	0.50
	BETA	67.81	71.52	64.62	60.86	76.20
10 %	UP	5225.00	1450.00	277.80	94.70	41.90
	DOWN	134.50	33.80	6.70	2.40	0.90
	BETA	38.85	42.90	41.46	39.46	46.56
20 %	UP	5339.00	1465.00	269.80	94.70	41.90
	DOWN	134.50	33.80	6.70	2.40	0.90
	BETA	39.70	43.34	40.27	39.46	46.56
40 %	UP	5433.00	1477.00	271.30	88.80	38.10
	DOWN	272.40	67.30	13.10	5.20	2.20
	BETA	19.94	21.95	20.71	17.08	17.32
80 %	UP	5461.00	1507.00	278.70	86.50	32.30
	DOWN	398.60	105.10	22.00	0.00	0.00
	BETA	13.70	14.34	12.67	*********	*********
AVERAGE BETA		30.60	33.15	30.87	40.04	45.46
MINIMUM BETA		13.70	14.34	12.67	17.08	17.32

ACFTD CAPACITY(GRAMS): APPARENT 12.2, RETAINED 11.9

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FILTER ELEMENT MULTI-PASS-TEST REPORT SHEET

FILTER ID:MERADCOM/FPRC QC TYPE 1 DATE: 05MAY78
LAB: FPRC-OSU FPRC NO. 441 FLOW RATE: 75.0 LPM R.H.: 50.0 %
BUBBLE POINT:17.8CM WATER ELEMENT SIZE (LXD) 22.9 X 7.0 CM.

DIFFERENTIAL PRESSURES,KPA

TERMINAL:	275.9
HOUSING:	31.0
CLEAN ASSEMBLY:	51.7
CLEAN ELEMENT:	20.7
NET:	255.2
SYSTEM VOLUME(L):	18.8

INJECTION	INITIAL	FINAL	AVERAGE
FLOW (LPM)	0.160	0.161	0.160
GRAV.(MG/L)	4780.0	4723.0	4736.5

UPSTREAM GRAVIMETRIC LEVELS(MG/L): BASE 10.13, FINAL 11.00

% NET PRESS	2.5%	5%	10%	20%	40%	80%	100%
ASSEMBLY PRESS	58.1	64.5	77.2	102.8	153.8	255.9	306.9
GRAMS ADDED	4.1	5.7	7.3	8.7	9.8	10.9	11.4
TEST TIME,MIN.	5.4	7.5	9.6	11.4	12.9	14.3	15.1

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE POINT		> 5 MIC	> 10 MIC	> 20 MIC	> 30 MIC	> 40 MIC
INITIAL		8.30	1.30	0.20	0.00	0.00
2MIN	UP	5285.00	1380.00	235.90	71.30	29.00
	DOWN	67.60	15.80	2.90	0.80	0.27
	BETA	77.95	87.34	81.34	89.13	108.61
10 %	UP	5578.00	1422.00	241.50	73.80	30.50
	DOWN	105.80	35.50	5.83	1.90	0.70
	BETA	52.72	40.06	41.42	38.84	43.57
20 %	UP	5433.00	1386.00	230.00	68.60	27.50
	DOWN	218.20	51.50	9.17	2.90	1.37
	BETA	24.90	26.91	25.08	23.72	20.07
40 %	UP	5637.00	1477.00	256.90	79.50	35.30
	DOWN	311.80	73.10	13.30	3.87	1.50
	BETA	18.72	20.21	19.32	20.54	23.53
80 %	UP	5877.00	1531.00	263.10	83.50	36.10
	DOWN	963.40	229.70	40.00	0.00	0.00
	BETA	6.10	6.67	6.58	*********	*********
AVERAGE BETA		26.16	25.59	25.29	38.66	40.85
MINIMUM BETA		6.10	6.67	6.58	20.54	20.07

ACFTD CAPACITY(GRAMS): APPARENT 11.4, RETAINED 11.2

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FILTER ELEMENT MULTI-PASS-TEST REPORT SHEET

FILTER ID:MERADCOM/FPRC QC TYPE 1 DATE: 03MAY78
LAB: FPRC-OSU FPRC NO. 442 FLOW RATE: 75.0 LPM R.H.: 40.0 %
BUBBLE POINT:34.3CM WATER ELEMENT SIZE (LXD) 22.9 X 7.0 CM.

DIFFERENTIAL PRESSURES,KPA

TERMINAL:	275.9
HOUSING:	31.0
CLEAN ASSEMBLY:	53.8
CLEAN ELEMENT:	22.8
NET:	253.1
SYSTEM VOLUME(L):	18.8

INJECTION	INITIAL	FINAL	AVERAGE
FLOW (LPM)	0.164	0.146	0.155
GRAV.(MG/L)	4855.0	4805.0	4830.0

UPSTREAM GRAVIMETRIC LEVELS(MG/L): BASE 9.99, FINAL 177.00

% NET PRESS	2.5%	5%	10%	20%	40%	80%	100%
ASSEMBLY PRESS	60.1	66.4	79.1	104.4	155.0	256.3	306.9
GRAMS ADDED	4.2	6.5	8.6	11.4	11.4	11.4	11.4
TEST TIME,MIN.	5.6	8.6	11.5	15.3	15.3	15.3	15.3

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE POINT		> 5 MIC	> 10 MIC	> 20 MIC	> 30 MIC	> 40 MIC
INITIAL		*********	*********	*********	*********	*********
10 %	UP	4705.00	1410.00	256.30	87.92	34.13
	DOWN	115.00	31.43	5.27	1.93	0.67
	BETA	40.91	44.86	48.63	45.55	50.94
20 %	UP	6208.00	1806.00	319.00	102.30	46.33
	DOWN	1817.00	531.10	98.17	32.33	14.53
	BETA	3.42	3.40	3.25	3.16	3.19
40 %	UP	12255.00	3680.00	676.30	218.80	93.50
	DOWN	8456.00	2536.00	452.30	145.80	62.83
	BETA	1.45	1.45	1.50	1.50	1.49
80 %	UP	*********	*********	*********	*********	*********
	DOWN	*********	*********	*********	*********	*********
	BETA	1.00	1.00	1.00	1.00	1.00
AVERAGE BETA		11.06	11.22	10.97	10.61	10.88
MINIMUM BETA		1.00	1.00	1.00	1.00	1.00

ACFTD CAPACITY(GRAMS): APPARENT 11.4, RETAINED 7.7

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FILTER ELEMENT MULTI-PASS-TEST REPORT SHEET

FILTER ID:MERADCOM/FPRC QC TYPE 1 DATE: 08MAY78
LAB: FPRC-OSU FPRC NO. 443 FLOW RATE: 75.0 LPM R.H.: 50.0 %
BUBBLE POINT:12.7CM WATER ELEMENT SIZE (LXD) 22.9 X 7.0 CM.

DIFFERENTIAL PRESSURES,KPA

TERMINAL:	275.9
HOUSING:	31.0
CLEAN ASSEMBLY:	52.4
CLEAN ELEMENT:	21.4
NET:	254.5
SYSTEM VOLUME(L):	18.8

INJECTION	INITIAL	FINAL	AVERAGE
FLOW (LPM)	0.162	0.163	0.162
GRAV.(MG/L)	4760.0	4770.0	4765.0

UPSTREAM GRAVIMETRIC LEVELS(MG/L): BASE 10.32, FINAL 18.00

% NET PRESS	2.5%	5%	10%	20%	40%	80%	100%
ASSEMBLY PRESS	58.8	65.2	77.9	103.3	154.2	255.9	306.9
GRAMS ADDED	4.4	6.4	8.2	9.8	11.1	12.3	12.9
TEST TIME,MIN.	5.6	3.3	10.6	12.7	14.3	15.9	16.6

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE POINT		> 5 MIC	> 10 MIC	> 20 MIC	> 30 MIC	> 40 MIC
INITIAL		10.00	2.00	0.30	0.10	0.00
2MIN	UP	5342.00	1400.00	245.60	77.60	32.90
	DOWN	81.20	41.23	3.30	1.27	0.70
	BETA	65.79	33.96	74.42	61.10	47.00
10 %	UP	5457.00	1414.00	254.30	79.40	33.50
	DOWN	127.90	31.10	5.63	1.70	0.83
	BETA	42.67	45.47	45.17	46.71	40.36
20 %	UP	5478.00	1440.00	252.00	77.20	32.20
	DOWN	193.70	45.13	8.40	2.93	1.37
	BETA	28.28	31.91	30.00	26.35	23.50
40 %	UP	5819.00	1507.00	258.60	77.40	33.10
	DOWN	310.30	74.43	14.63	4.93	2.23
	BETA	18.75	20.25	17.68	15.70	14.84
80 %	UP	6039.00	1560.00	275.20	83.60	34.60
	DOWN	858.20	214.90	40.50	0.00	0.00
	BETA	7.04	7.35	6.80	**********	**********
AVERAGE BETA		25.92	26.26	26.16	39.81	35.04
MINIMUM BETA		7.04	7.35	6.80	15.70	14.84

ACFTD CAPACITY(GRAMS): APPARENT 12.9, RETAINED 12.5

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FILTER ELEMENT MULTI-PASS-TEST REPORT SHEET

FILTER ID:MERADCOM/FPRC QC TYPE 1 DATE: 05APR78
LAB: FPRC-OSU FPRC NO. 444 FLOW RATE: 75.0 LPM R.H.: 40.0 %
BUBBLE POINT:29.2CM WATER ELEMENT SIZE (LXD) 22.9 X 7.0 CM.

DIFFERENTIAL PRESSURES,KPA

TERMINAL:	275.9
HOUSING:	31.0
CLEAN ASSEMBLY:	50.3
CLEAN ELEMENT:	19.3
NET:	256.6
SYSTEM VOLUME(L):	18.8

INJECTION	INITIAL	FINAL	AVERAGE
FLOW (LPM)	0.164	0.163	0.163
GRAV.(MG/L)	4830.0	4800.0	4815.0

UPSTREAM GRAVIMETRIC LEVELS(MG/L): BASE 10.49, FINAL 22.00

% NET PRESS	2.5%	5%	10%	20%	40%	80%	100%
ASSEMBLY PRESS	56.8	63.2	76.0	101.7	153.0	255.6	306.9
GRAMS ADDED	4.7	6.4	7.9	9.4	10.9	10.9	10.9
TEST TIME,MIN.	6.0	8.1	10.0	11.9	13.9	13.9	13.9

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE POINT		> 5 MIC	> 10 MIC	> 20 MIC	> 30 MIC	> 40 MIC
INITIAL		11.60	2.00	0.40	0.20	0.10
10 %	UP	4939.00	1453.00	257.10	81.20	34.20
	DOWN	91.27	26.07	4.73	1.73	0.80
	BETA	54.11	55.96	54.36	46.94	42.75
20 %	UP	5050.00	1507.00	267.90	88.33	40.20
	DOWN	156.90	64.23	11.17	3.83	1.77
	BETA	31.78	23.46	23.98	23.06	22.71
40 %	UP	5259.00	1562.00	283.80	93.40	40.47
	DOWN	353.50	100.00	17.60	6.43	1.77
	BETA	14.88	15.62	16.12	14.53	22.86
80 %	UP	5953.00	1744.00	303.70	97.87	43.33
	DOWN	1254.00	356.60	65.83	23.70	10.63
	BETA	4.75	4.89	4.61	4.13	4.08
AVERAGE BETA		36.87	36.02	36.06	32.23	34.38
MINIMUM BETA		4.75	4.89	4.61	4.13	4.08

AEFTD CAPACITY(GRAMS): APPARENT 10.9, RETAINED 10.5

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FILTER ELEMENT MULTI-PASS-TEST REPORT SHEET

FILTER ID:MERADCOM/FPRC QC TYPE 1 DATE: 08MAY78
LAB: FPRC-OSU FPRC NO. 445 FLOW RATE: 75.0 LPM R.H.: 55.0 %
BUBBLE POINT:21.6CM WATER ELEMENT SIZE (LXD) 22.9 X 7.0 CM.
DIFFERENTIAL PRESSURES,KPA

TERMINAL:	275.9
HOUSING:	31.0
CLEAN ASSEMBLY:	51.7
CLEAN ELEMENT:	20.7
NET:	255.2
SYSTEM VOLUME(L):	18.8

INJECTION	INITIAL	FINAL	AVERAGE
FLOW (LPM)	0.160	0.162	0.161
GRAV.(MG/L)	4705.01	4740.01	4722.51

UPSTREAM GRAVIMETRIC LEVELS(MG/L): BASE 10.14, FINAL 169.00

% NET PRESS	2.5%	5%	10%	20%	40%	80%	100%
ASSEMBLY PRESS	58.1	64.5	77.2	102.6	153.8	255.9	306.9
GRAMS ADDED	4.2	6.0	7.6	9.3	11.1	11.1	11.1
TEST TIME,MIN.	5.5	7.9	10.0	12.2	14.6	14.6	14.6

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE POINT		> 5 MIC	> 10 MIC	> 20 MIC	> 30 MIC	> 40 MIC
INITIAL		8.90	3.00	1.20	0.80	0.50
10 %	UP	5071.00	1414.00	228.50	67.20	24.87
	DOWN	84.60	22.70	4.33	1.53	0.73
	BETA	59.94	62.29	52.77	43.92	34.07
20 %	UP	5324.00	1148.00	242.40	73.13	30.07
	DOWN	258.30	68.60	12.27	4.03	2.07
	BETA	20.61	16.73	19.76	18.15	14.53
40 %	UP	5544.00	1538.00	253.50	81.83	38.00
	DOWN	570.20	153.20	27.17	9.30	4.57
	BETA	9.72	10.04	9.33	8.80	8.32
80 %	UP	8093.00	2224.00	371.00	115.30	49.83
	DOWN	4019.00	1132.00	191.70	62.40	28.97
	BETA	2.01	1.96	1.94	1.85	1.72
AVERAGE BETA		27.88	27.33	26.00	23.35	19.81
MINIMUM BETA		2.01	1.96	1.94	1.85	1.72

ACFTD CAPACITY(GRAMS): APPARENT 11.1, RETAINED 7.6

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FILTER ELEMENT MULTI-PASS-TEST REPORT SHEET

FILTER ID:MERADCOM/FPRC QC TYPE 2 DATE: 25APR78
LAB: FPRC-OSU FPRC NO. 446 FLOW RATE: 75.0 LPM R.H.: 40.0 %
BUBBLE POINT:11.4CM WATER ELEMENT SIZE (LXD) 17.8 X 11.4 CM.

DIFFERENTIAL PRESSURES,KPA

TERMINAL:	275.9
HOUSING:	51.7
CLEAN ASSEMBLY:	62.1
CLEAN ELEMENT:	10.3
NET:	265.5
SYSTEM VOLUME(L):	18.8

INJECTION	INITIAL	FINAL	AVERAGE
FLOW (LPM)	0.250	0.249	0.249
GRAV.(MG/L)	3104.0	3110.0	3107.0

UPSTREAM GRAVIMETRIC LEVELS(MG/L): BASE 10.33, FINAL 87.00

% NET PRESS	2.5%	5%	10%	20%	40%	80%	100%
ASSEMBLY PRESS	68.7	75.2	88.3	115.2	168.3	274.5	327.6
GRAMS ADDED	18.2	27.4	32.5	33.0	38.9	41.5	41.9
TEST TIME,MIN.	23.5	35.3	41.9	42.6	50.2	53.5	54.0

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE POINT		> 5 MIC	> 10 MIC	> 20 MIC	> 30 MIC	> 40 MIC
INITIAL		**********	**********	**********	**********	**********
2MIN	UP	12600.00	2084.00	308.00	103.30	44.00
	DOWN	8218.00	609.00	17.33	3.00	1.00
	BETA	1.53	3.42	17.77	34.43	44.00
10 %	UP	44900.00	4499.00	322.00	93.33	36.67
	DOWN	41220.00	2979.00	38.67	6.00	1.33
	BETA	1.09	1.51	8.33	15.56	27.57
20 %	UP	55710.00	5845.00	316.70	90.67	41.33
	DOWN	46950.00	3955.00	54.00	2.00	0.00
	BETA	1.19	1.48	5.86	45.33	**********
40 %	UP	56820.00	6960.00	326.70	87.33	35.33
	DOWN	55160.00	5557.00	70.00	5.33	1.33
	BETA	1.03	1.25	4.67	16.38	26.56
80 %	UP	58830.00	7349.00	376.00	90.00	30.67
	DOWN	48470.00	5480.00	90.00	0.00	0.00
	BETA	1.21	1.34	4.18	**********	**********
AVERAGE BETA		1.10	1.49	7.19	17.24	30.08
MINIMUM BETA		1.03	1.25	4.18	15.56	26.56

ACFTD CAPACITY(GRAMS): APPARENT 41.9, RETAINED 39.2

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FILTER ELEMENT MULTI-PASS-TEST REPORT SHEET

FILTER ID:MERADCOM/FPRC QC TYPE 2 DATE: 21APR78
LAB: FPRC-OSU FPRC NO. 447 FLOW RATE: 75.0 LPM R.H.: 40.0 %
BUBBLE POINT: 6.9CM WATER ELEMENT SIZE (LXD) 17.8 X 11.4 CM.

DIFFERENTIAL PRESSURES,KPA

TERMINAL:	275.9
HOUSING:	51.7
CLEAN ASSEMBLY:	62.1
CLEAN ELEMENT:	10.3
NET:	265.5

SYSTEM VOLUME(L): 18.8

INJECTION	INITIAL	FINAL	AVERAGE
FLOW (LPM)	0.246	0.248	0.247
GRAV.(MG/L)	3160.0	3180.0	3170.0

UPSTREAM GRAVIMETRIC LEVELS(MG/L): BASE 10.44, FINAL 72.00

% NET PRESS	2.5%	5%	10%	20%	40%	80%	100%
ASSEMBLY PRESS	68.7	75.2	88.3	115.2	168.3	274.5	327.6
GRAMS ADDED	12.2	17.8	22.3	26.2	29.3	31.3	33.0
TEST TIME,MIN.	15.6	22.7	28.5	33.5	37.4	40.0	42.2

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE POINT		> 5 MIC	> 10 MIC	> 20 MIC	> 30 MIC	> 40 MIC
INITIAL		**********	**********	**********	**********	**********
2MIN	UP	9961.00	1866.00	260.30	79.00	32.00
	DOWN	3908.00	326.00	17.00	5.00	3.00
	BETA	2.55	5.72	15.31	15.80	10.67
10 %	UP	33563.00	3497.00	282.00	85.70	27.70
	DOWN	25960.00	1813.00	33.00	12.30	6.00
	BETA	1.29	1.93	8.55	6.97	4.62
20 %	UP	33880.00	3479.00	301.00	90.00	39.70
	DOWN	31180.00	2482.00	31.00	2.00	0.33
	BETA	1.09	1.40	9.71	45.00	120.30
40 %	UP	36970.00	4493.00	293.70	91.30	38.30
	DOWN	33540.00	3263.00	39.30	3.00	1.33
	BETA	1.10	1.38	7.47	30.43	28.80
80 %	UP	38400.00	5960.00	434.00	126.00	53.00
	DOWN	33910.00	4361.00	130.00	0.00	0.00
	BETA	1.13	1.37	3.34	**********	**********
AVERAGE BETA		1.26	1.79	7.33	9.59	6.55
MINIMUM BETA		1.09	1.37	3.34	6.97	4.62

ACFTD CAPACITY(GRAMS): APPARENT 33.0, RETAINED 31.0

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FILTER ELEMENT MULTI-PASS-TEST REPORT SHEET

FILTER ID:MERADCOM/FPRC QC TYPE 2 DATE: 21APR78
LAB: FPRC-OSU FPRC NO. 448 FLOW RATE: 75.0 LPM R.H.: 40.0 %
BUBBLE POINT: 7.0CM WATER ELEMENT SIZE (LXD) 17.8 X 11.4 CM.

DIFFERENTIAL PRESSURES,KPA

TERMINAL:	275.9
HOUSING:	51.7
CLEAN ASSEMBLY:	62.1
CLEAN ELEMENT:	10.3
NET:	265.5
SYSTEM VOLUME(L):	18.8

INJECTION	INITIAL	FINAL	AVERAGE
FLOW (LPM)	0.243	0.240	0.241
GRAV.(MG/L)	3172.0	3152.0	3162.0

UPSTREAM GRAVIMETRIC LEVELS(MG/L): BASE 10.18, FINAL 79.00

% NET PRESS	2.5%	5%	10%	20%	40%	80%	100%
ASSEMBLY PRESS	68.7	71.0	88.6	115.2	168.3	274.5	327.6
GRAMS ADDED	10.8	16.8	20.6	24.4	27.9	31.3	32.8
TEST TIME,MIN.	14.2	22.0	27.0	32.0	36.5	41.0	43.0

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE POINT		> 5 MIC	> 10 MIC	> 20 MIC	> 30 MIC	> 40 MIC
INITIAL		*********	*********	*********	*********	*********
2MIN	UP	8795.00	1763.00	271.00	94.70	44.00
	DOWN	4015.00	328.00	6.33	2.00	0.33
	BETA	2.19	5.38	42.81	47.35	133.33
10 %	UP	29834.00	3290.00	285.00	86.30	46.30
	DOWN	25690.00	1831.00	24.00	3.67	1.67
	BETA	1.16	1.80	11.88	23.51	27.72
20 %	UP	44065.00	35673.00	372.00	113.30	50.30
	DOWN	37450.00	4114.00	117.70	23.30	9.33
	BETA	1.18	8.67	3.16	4.86	5.39
40 %	UP	34335.00	3940.00	298.00	87.00	40.30
	DOWN	31210.00	2661.00	43.70	9.00	4.30
	BETA	1.10	1.48	6.82	9.67	9.37
60 %	UP	41033.00	6828.00	541.00	163.00	66.30
	DOWN	38056.00	5800.00	343.00	0.00	0.00
	BETA	1.08	1.18	1.58	*********	*********
AVERAGE BETA		1.17	1.89	5.66	15.77	17.64
MINIMUM BETA		1.08	1.18	1.58	4.86	5.39

ACFTD CAPACITY(GRAMS): APPARENT 32.8, RETAINED 30.6

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FILTER ELEMENT MULTI-PASS-TEST REPORT SHEET

FILTER ID:MERADCOM/FPRC QC TYPE 3 DATE: 05MAY78
LAB: FPRC-GSU FPRC NO. 449 FLOW RATE: 75.0 LPM R.H.: 50.0 %
BUBBLE POINT:14.0CM WATER ELEMENT SIZE (LXD) 23.5 X 10.2 CM.

DIFFERENTIAL PRESSURES,KPA

TERMINAL:	275.9
HOUSING:	15.2
CLEAN ASSEMBLY:	22.1
CLEAN ELEMENT:	6.9
NET:	269.0

SYSTEM VOLUME(L): 18.8

INJECTION	INITIAL	FINAL	AVERAGE
FLOW (LPM)	0.157	0.144	0.150
GRAV.(MG/L)	4767.0	4773.0	4770.0

UPSTREAM GRAVIMETRIC LEVELS(MG/L): BASE 9.57, FINAL 139.00

% NET PRESS	2.5%	5%	10%	20%	40%	80%	100%
ASSEMBLY PRESS	28.8	35.5	49.0	75.9	129.7	237.2	291.0
GRAMS ADDED	15.9	23.1	28.7	32.4	35.1	37.0	37.6
TEST TIME,MIN.	22.1	32.2	40.0	45.1	48.9	51.6	52.4

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE POINT		> 5 MIC	> 10 MIC	> 15 MIC	> 20 MIC	> 25 MIC
INITIAL		19.50	3.40	1.00	0.50	0.20
2MIN	UP	8093.00	1458.00	544.30	260.70	138.30
	DOWN	2870.00	83.00	11.33	4.67	2.00
	BETA	2.82	17.57	48.04	55.82	69.15
10 %	UP	25970.00	1780.00	474.00	215.30	105.00
	DOWN	21973.00	486.00	24.00	4.33	0.67
	BETA	1.18	3.66	19.75	49.72	156.72
20 %	UP	30600.00	1826.00	450.30	210.30	114.70
	DOWN	26440.00	689.00	35.00	2.67	0.33
	BETA	1.16	2.65	12.87	78.76	347.58
40 %	UP	38137.00	2150.00	406.00	177.70	93.30
	DOWN	33090.00	1075.00	49.00	6.33	0.33
	BETA	1.15	2.00	8.29	28.07	282.73
80 %	UP	52290.00	3397.00	600.00	238.70	124.70
	DOWN	46560.00	2040.00	124.70	0.00	0.00
	BETA	1.12	1.66	4.81	*********	*********
AVERAGE BETA		1.20	3.24	15.26	51.59	172.28
MINIMUM BETA		1.12	1.66	4.81	28.07	69.15

ACFTD CAPACITY(GRAMS): APPARENT 37.6, RETAINED 34.0

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FILTER ELEMENT MULTI-PASS-TEST REPORT SHEET

FILTER ID:MERADCOM/FPRC GC TYPE 3 DATE: 09MAY78
LAB: FPRC-OSU FPRC NO. 450 FLOW RATE: 75.0 LPM R.H.: 50.0 %
BUBBLE POINT: 8.9CM WATER ELEMENT SIZE (LXD) 23.5 X 10.2 CM.

DIFFERENTIAL PRESSURES,KPA

TERMINAL:	275.9
HOUSING:	15.2
CLEAN ASSEMBLY:	22.1
CLEAN ELEMENT:	6.9
NET:	269.0

SYSTEM VOLUME(L): 18.8

INJECTION	INITIAL	FINAL	AVERAGE
FLOW (LPM)	0.158	0.158	0.158
GRAV.(MG/L)	4800.0	4768.0	4784.0

UPSTREAM GRAVIMETRIC LEVELS(MG/L): BASE 10.06, FINAL 160.00

% NET PRESS	2.5%	5%	10%	20%	40%	80%	100%
ASSEMBLY PRESS	28.8	35.5	49.0	75.9	129.7	237.2	291.0
GRAMS ADDED	16.6	24.5	30.5	34.0	35.8	37.3	37.9
TEST TIME,MIN.	22.0	32.4	40.4	45.0	47.3	49.4	50.2

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE POINT		> 5 MIC	> 10 MIC	> 15 MIC	> 20 MIC	> 25 MIC
INITIAL		23.40	10.20	8.00	5.90	4.10
2MIN	UP	8155.00	1451.00	523.30	255.70	138.70
	DOWN	2364.00	77.33	8.33	3.00	1.00
	BETA	3.45	18.76	62.82	85.23	138.70
10 %	UP	28528.00	1993.00	540.70	252.30	125.00
	DOWN	24095.00	509.00	28.00	5.67	1.67
	BETA	1.18	3.27	19.31	44.50	74.85
20 %	UP	37080.00	2396.00	564.00	251.70	138.70
	DOWN	32060.00	986.30	53.00	8.33	1.67
	BETA	1.16	2.43	10.64	30.22	83.05
40 %	UP	47690.00	3126.00	633.00	268.70	134.30
	DOWN	42000.00	1600.00	99.00	10.67	2.00
	BETA	1.14	1.95	6.39	25.18	67.15
80 %	UP	55274.00	3746.00	649.00	255.70	133.30
	DOWN	49580.00	2314.00	129.30	0.00	0.00
	BETA	1.11	1.62	5.02	*********	*********
AVERAGE BETA		1.21	3.05	15.32	43.67	79.97
MINIMUM BETA		1.11	1.62	5.02	25.18	67.15

ACFTO CAPACITY(GRAMS): APPARENT 37.9, RETAINED 33.8

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FILTER ELEMENT MULTI-PASS-TEST REPORT SHEET

FILTER ID:MERADCOM/FPRC QC TYPE 3 DATE: 09MAY78
LAB: FPRC-OSU FPRC NO. 451 FLOW RATE: 75.0 LPM R.H.: 50.0 %
BUBBLE POINT:19.0CM WATER ELEMENT SIZE (LXD) 23.5 X 10.2 CM.

DIFFERENTIAL PRESSURES,KPA

TERMINAL:	275.9
HOUSING:	15.2
CLEAN ASSEMBLY:	22.1
CLEAN ELEMENT:	6.9
NET:	269.0

SYSTEM VOLUME(L): 18.8

INJECTION	INITIAL	FINAL	AVERAGE
FLOW (LPM)	0.161	0.162	0.161
GRAV.(MG/L)	4512.01	4500.01	4506.01

UPSTREAM GRAVIMETRIC LEVELS(MG/L): BASE 9.70, FINAL 138.00

% NET PRESS	2.5%	5%	10%	20%	40%	80%	100%
ASSEMBLY PRESS	28.81	35.51	49.01	75.91	129.71	237.21	291.01
GRAMS ADDED	15.61	24.51	29.81	32.81	34.91	36.51	36.91
TEST TIME,MIN.	21.51	33.71	41.01	45.11	47.91	50.11	50.71

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE POINT		> 5 MIC	> 10 MIC	> 15 MIC	> 20 MIC	> 25 MIC
INITIAL		22.201	4.401	1.701	0.601	0.501
2MIN	UP	8676.001	1428.001	497.001	241.301	127.701
	DOWN	2733.001	80.331	7.671	3.671	1.001
	BETA	3.171	17.781	64.801	65.751	127.701
10 %	UP	34861.001	2225.001	533.001	235.701	121.001
	DOWN	27212.001	732.701	41.331	7.001	3.331
	BETA	1.281	3.041	12.901	33.671	36.341
20 %	UP	43645.001	2606.001	530.001	228.001	120.701
	DOWN	39361.001	1257.001	56.001	6.001	1.331
	BETA	1.111	2.071	9.461	38.001	90.751
40 %	UP	55400.001	3428.001	630.001	253.601	131.701
	DOWN	49044.001	1927.001	89.671	8.671	0.671
	BETA	1.131	1.781	7.031	29.251	196.571
80 %	UP	64610.001	4414.001	693.701	250.301	116.001
	DOWN	59400.001	2983.001	182.301	0.001	0.001
	BETA	1.091	1.481	3.811	*********	*********
AVERAGE BETA		1.281	2.761	11.221	35.981	43.631
MINIMUM BETA		1.091	1.481	3.811	29.251	36.341

ACFTD CAPACITY(GRAMS): APPARENT 36.9, RETAINED 33.3

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FILTER ELEMENT MULTI-PASS-TEST REPORT SHEET

FILTER ID:MERADCOM/FPRC QC TYPE 3 DATE: 09MAY78
LAB: FPRC-OSU FPRC NO. 452 FLOW RATE: 75.0 LPM R.H.: 50.0 %
BUBBLE POINT: 5.1CM WATER ELEMENT SIZE (LXD) 23.5 X 10.2 CM.

DIFFERENTIAL PRESSURES,KPA

TERMINAL:	275.9
HOUSING:	15.2
CLEAN ASSEMBLY:	22.1
CLEAN ELEMENT:	6.9
NET:	269.0
SYSTEM VOLUME(L):	18.8

INJECTION	INITIAL	FINAL	AVERAGE
FLOW (LPM)	0.165	0.161	0.163
GRAV.(MG/L)	4504.0	4512.0	4508.0

UPSTREAM GRAVIMETRIC LEVELS(MG/L): BASE 9.79, FINAL 142.00

% NET PRESS	2.5%	5%	10%	20%	40%	80%	100%
ASSEMBLY PRESS	28.8	35.5	49.0	75.9	129.7	237.2	291.0
GRAMS ADDED	17.6	24.8	29.0	31.0	33.4	36.2	36.6
TEST TIME,MIN.	24.0	33.8	39.5	42.2	45.5	49.2	50.1

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE POINT		> 5 MIC	> 10 MIC	> 20 MIC	> 30 MIC	> 40 MIC
INITIAL		*********	*********	*********	*********	*********
2MIN	UP	87830.00	1524.00	261.70	75.00	26.67
	DOWN	2585.00	79.67	0.67	0.00	0.00
	BETA	33.98	19.13	390.60	*********	*********
10 %	UP	37430.00	2292.00	243.70	78.67	35.33
	DOWN	29730.00	784.70	4.67	0.67	0.67
	BETA	1.26	2.92	52.18	117.42	52.73
20 %	UP	46040.00	2699.00	243.70	72.67	29.00
	DOWN	38252.00	1115.00	6.33	1.00	0.33
	BETA	1.20	2.42	38.50	72.67	87.88
40 %	UP	53463.00	3129.00	241.30	69.67	27.67
	DOWN	50010.00	1731.00	8.00	0.67	0.00
	BETA	1.07	1.81	30.16	103.99	*********
80 %	UP	65160.00	4245.00	281.00	78.00	28.00
	DOWN	60555.00	2653.00	9.33	0.00	0.00
	BETA	1.09	1.60	30.12	*********	*********
AVERAGE BETA		1.28	2.70	47.95	126.72	66.32
MINIMUM BETA		1.07	1.60	30.12	72.67	52.73

ACFTD CAPACITY(GRAMS): APPARENT 36.8, RETAINED 33.1